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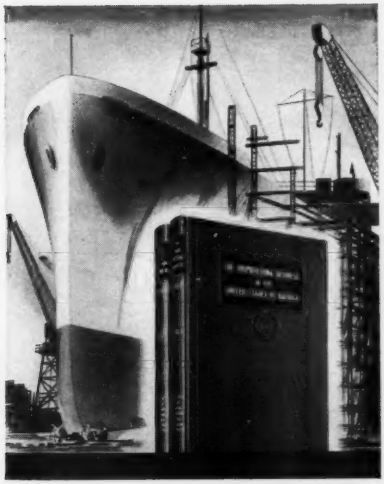

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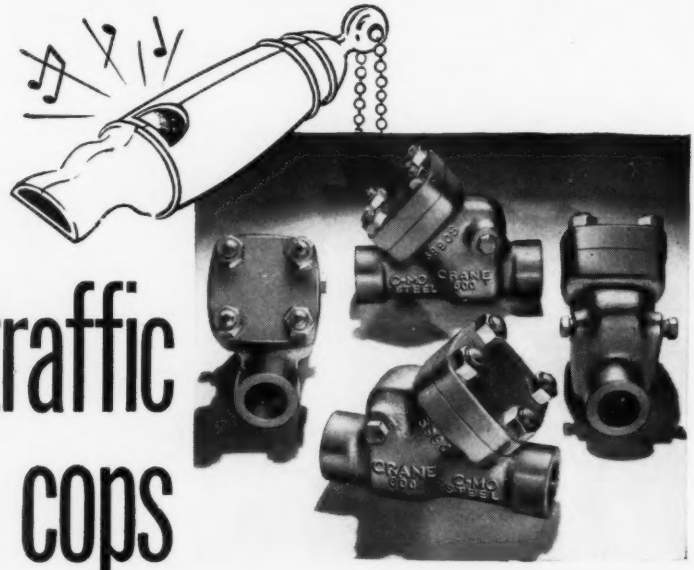


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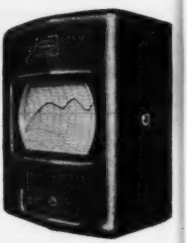

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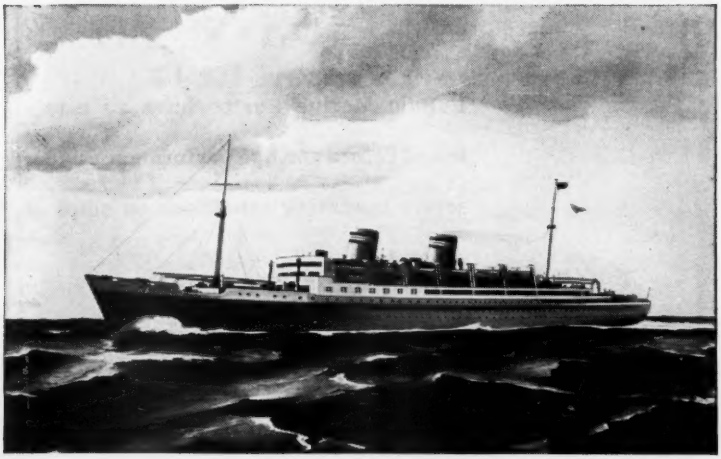

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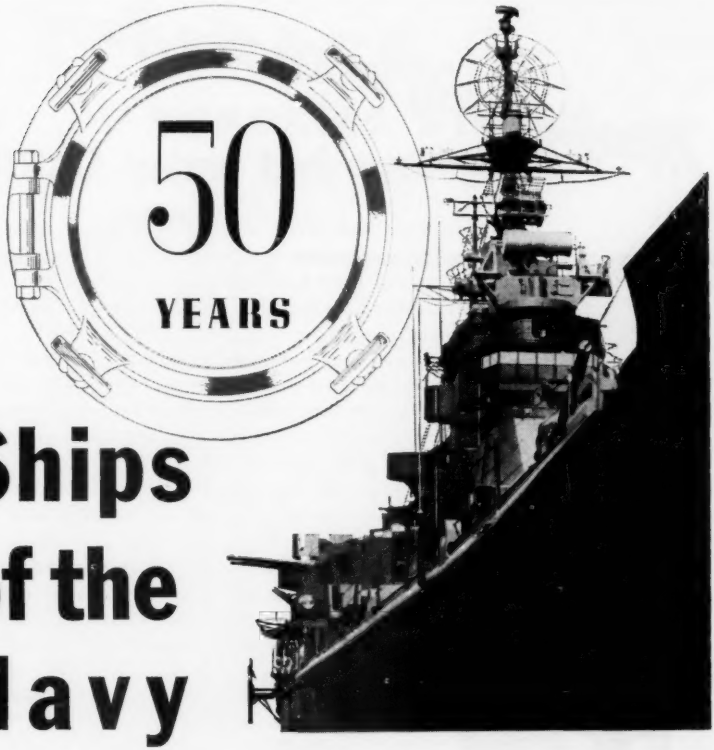


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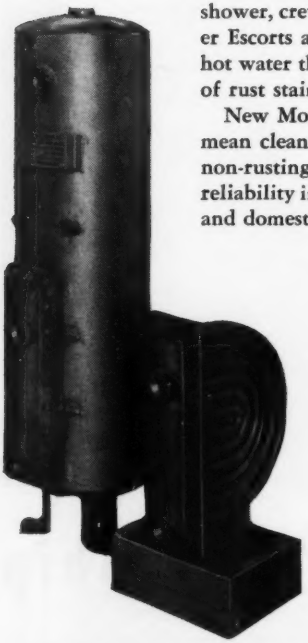

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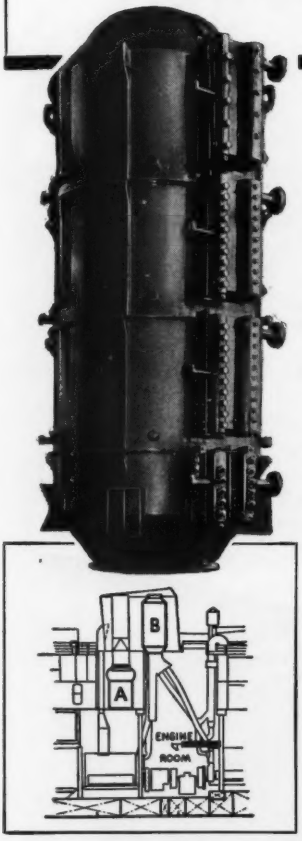


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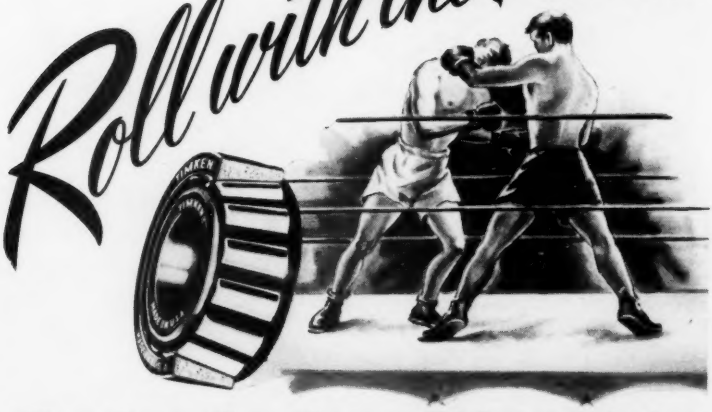


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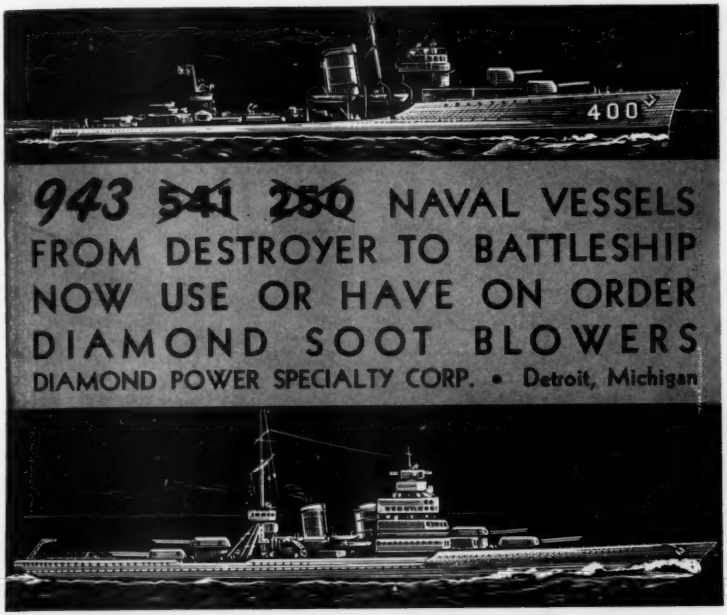


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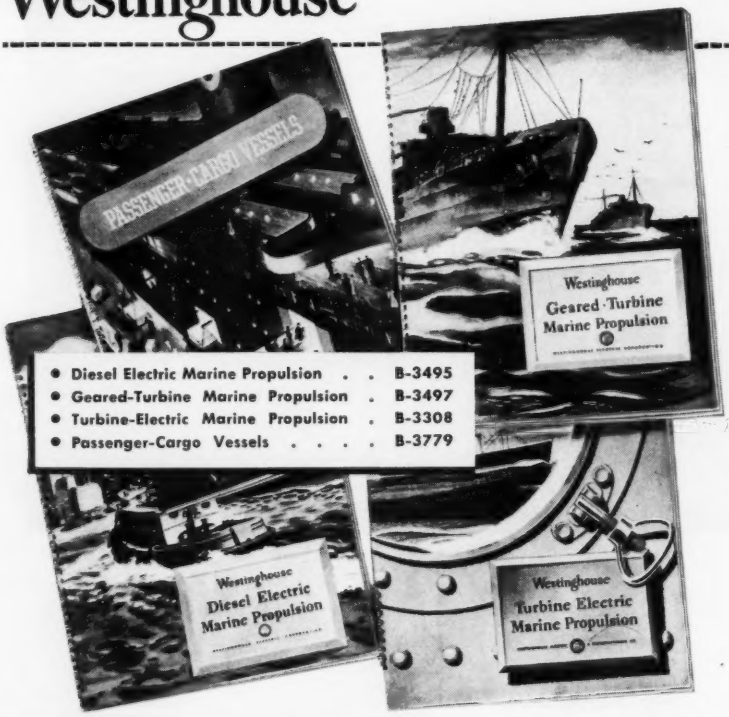


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OF

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# PART ONE—ORIGINAL ARTICLES

The articles which the Editor, acting under the supervision of the Council, has accepted for publication in Part One are the original works of the authors.

They are offered in furtherance of the Society's raison d'etre: to further the advancement of naval engineering. Discussion of any article is invited for publication in a subsequent issue.

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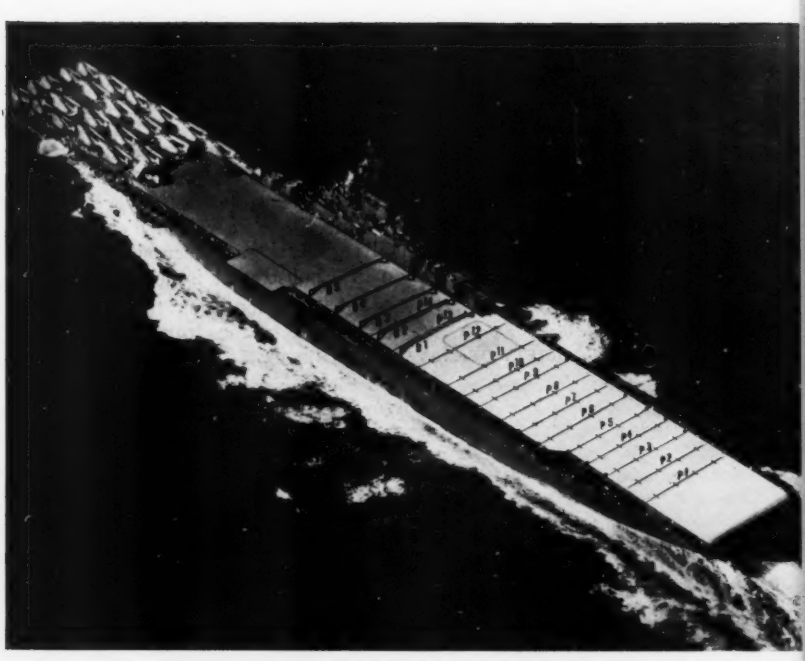


AIR CRAFT CARRIER ARRESTING GEAR.

# CARRIER ARRESTING GEAR—How one of the mechanical elements which contributed to winning the war was developed.

BY JOSEPH C. TERRY.

THE AUTHOR.

JOSEPH C. TERRY received his B.S. in Mechanical Engineering from the University of New Hampshire in 1931. His early experience was in automotive, civil, and refrigeration engineering, the latter with the York Ice Machinery Corp., York, Pa., for six years prior to World War II. In March 1942, he received appointment as an engineer in the arresting gear section of the Bureau of Aeronautics which association was continued as a commissioned officer in the Naval Reserve. Upon release from active duty, he returned to his former civilian position where he is presently engaged in administering the arresting gear and barrier development program.

"Arresting Gear", as the name implies, is the equipment installed in aircraft carriers to stop all forward motion of aircraft when landing on the flight decks of such vessels. As used today, it not only includes that equipment by which an airplane is normally arrested, but also the emergency barriers which separate the landing and the parking areas. Any medium or device which acts to retard the motion of a body is in a sense "Arresting Gear". The development of the wheel brake was perhaps the earliest approach to an energy-absorbing device which could be called "Arresting Gear". However, the ability of brakes, installed *on the moving body*, to transform kinetic energy into heat at the high rate demanded by the short stopping distance on carrier flight decks, is limited by the coefficient of friction between the moving and static body. It became necessary, therefore, to conceive means whereby a retarding load could be applied to the airplane in a more positive manner.

*Early History.*

1. (a) The history of arresting gear began in 1911 when Eugene Ely landed a 1000 lb. airplane on a special platform erected on a gun turret aboard the cruiser U.S.S. *Pennsylvania*. The platform, 120 ft. long and 32 ft. wide, was equipped with a series of 22 athwartship lines spaced about 3 ft. apart and secured at each end to a 50 lb. sandbag lying on the deck. Each line was raised off the deck about 12″ by several supports and the airplane was equipped with three pairs of crude hooks which caught the raised lines. All the basic essentials to arresting gear were present at that time. However, the continuing development of those essentials has made it possible for Naval forces to utilize the airplane as a potent weapon of war.

(b) Other Naval powers visualized the possibilities of Naval Aviation at about the same time. In 1912, Pegoud, the French aviator, conceived and installed a suspended cable system wherein the pilot

endeavored to engage a hook, installed in and above the airplane, on a longitudinal cable suspended above the deck. The disadvantages of this system compared with the athwartship cable system are quite obvious and as a result the French soon resorted to the athwartship cable system.

In August 1917, the British began on H.M.S. *Furious* with no arresting gear as such. Aircraft were equipped with skids and provisions to prevent nosing over. The resulting friction was considered adequate to stop the airplane. For emergency purposes, a boom was suspended above the forward part of the deck and 2″ diameter manila lines were trailed from this boom to deck level. Any airplane over-running the landing area would become entangled in the suspended lines and stopped by causing work to be done on the aircraft structure. The *Furious* arrangement gave way to a fore and aft cable system with inclined ramp installed in H.M.S. *Argus*. A series of steel hooks were installed on the horizontal landing gear axles, which engaged fore and aft cables secured at each end to the deck but supported about 12″ above it. These cables were spaced about 12″ apart. Upon reaching the inclined ramp, the airplane caused an up-haul on the secured cables. The amount of work done was a function of the strain and resulting tension produced in the cables. This gear was later removed and weighted athwartship cables were substituted.

*Definitions.*

Before going further it will be beneficial to define a few arresting gear terms which must be used in tracing the development of arresting gear components:

ARRESTING UNIT OR ENGINE—the energy absorber.

DECK PENDANT—a replaceable section of wire rope inserted in the wire rope system of the energy absorber and engaged by the arresting hook installed on the airplane.

PURCHASE CABLE—The major portion of the wire rope system, similar to the rope on a block and tackle, which carries the developed retarding loads.

YIELDING ELEMENTS — WIRE SUPPORTS — Those devices employed to raise the deck pendant clear of the deck.

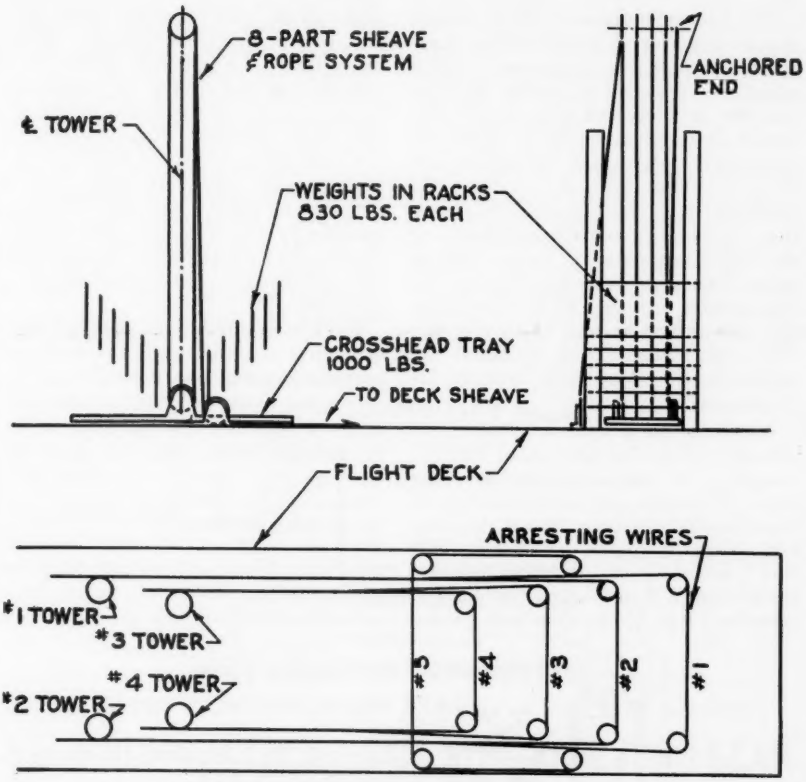
SHEAVES — Grooved wheels similar to pulleys which carry the wire rope and change its direction.

BARRIERS—Wire rope fences erected about mid-way of the flight deck to engage the airplane on its forward portion—propeller, engine, and landing gear—and prevent further passage of the airplane into aircraft parked on the forward part of the flight deck (For emergency use).

*Energy Absorbers.*

The conversion of the collier *Jupiter*, rechristened the U.S.S. *Langley* (CV-1), upon completion in 1922 provided the first actual aircraft carrier. She was equipped with a *"weight-type"* energy absorber (converter) (Figure 1) comprised of two sets of towers erected at the deck edge and a series of weights for each tower. Upon engagement of an athwartship line, which was reeved over sheaves in the tower and on a moving (rising) crosshead, a series of weights was lifted in sequence. By varying the area of the rising crosshead tray a greater number of weights would be lifted and different amounts of energies could be absorbed.

The first bonafide arresting engine appeared in 1927 when the "Norden gear" or Mark 2 brake was installed in the U.S.S. *Lexington* and *Saratoga* (CV-2 and -3) (Figure 2). The Mark 2 brake, designed by C. L. Norden and T. H. Barth, was a band type brake by which rotary motion of a cable drum was resisted. Band tension was provided by hydraulic pressure developed in a positive displacement pump driven from the cable drum shaft. Pump pressure acting on a hydraulic piston applied take-up force to the brake band through a lever system. Control over maximum pump pressure was accomplished through an air-loaded by-pass valve. Retraction to battery position was accomplished by an electric motor driving through a cone clutch and worm gearing. Static tension in the deck pendants was provided by

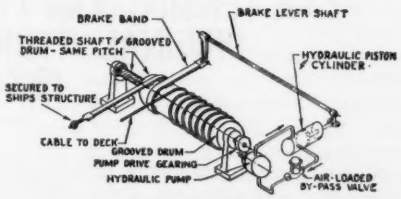



**WEIGHT TYPE ARRESTING GEAR**
**U.S.S. LANGLEY (CV-1) ORIGINAL**
**1922**

FIGURE 1.

compressed springs. The maximum capacity of this gear was a 10,000# airplane engaging at 60 mph and resulted in a maximum deceleration of 2 "g" for 100 ft. run.

There were two (2) brakes for each arresting wire, one brake at each end. Such an arrangement proved to be a disadvantage, since, if the airplane did not pick up the deck pendant mid-way between deck sheaves, each cable drum would pay out only on demand and extra cable had to be wound on each drum to accommodate extreme off-center conditions.



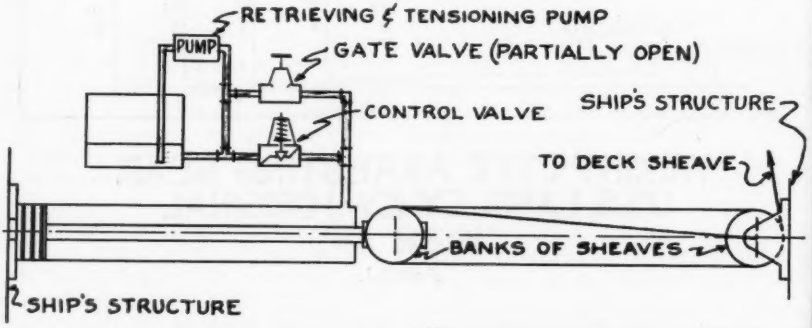

**MK.2 (NORDEN) BRAKE**
**U.S.S. LEXINGTON & SARATOGA**
**1927**

FIGURE 2.

The hydraulic pressure for the brakes was set alike for all brakes from a single control station, but additional braking effort was provided in brakes near the barriers by increased area in the air-loaded by-pass valve piston. Other disadvantages were the necessity for frequent brake block replacement, dependence upon ship's power for retraction (retrieving) and the complexity of the deck pendant tension spring arrangement. This gear was used with a deck arrangement utilizing both a fore and aft cable system as well as an athwartship cable system (deck pendants) described under "Wire Rope System Arrangements".

The need for a lighter, simpler, and smoother brake brought about the development of hydraulic type arresting gear. The design was undertaken by the Experimental Division, Naval Air Station, Hampton Roads, Virginia in 1928 and resulted in an experimental unit comprising a tension-ram, piston and cylinder (Figure 3). One end of the cylinder was secured to the foundation. The piston was pulled out of the cylinder by a wire rope and sheave system similar to a block and tackle with one set of sheaves mounted on the end of the piston rod and the other secured to the foundation. Fluid trapped between the piston and piston rod packing was forced through a hydraulic pressure control valve into a sump by the action of the piston. One end of the wire rope purchase was secured to the foundation and the other connected to a deck pendant. Two of these units were used for each deck pendant. The deck pendants were returned to battery and tensioned by an electrically driven pump which returned fluid to the cylinder displacing the piston and maintained hydraulic pressure in the cylinder.

The tension-ram design required heavy structural foundations to withstand the loads imposed by the unit. Perfect seal around the piston was difficult with piston rings of that day and the inherent disadvantages of two units for each wire were present. Further development to overcome these difficulties resulted in the Mark 3 compression-ram type wherein the ram displaced its own volume of fluid when forced *into* the cylinder by the action of two banks of sheaves being pulled together by the purchase cable reeved around them. Retrieving of the pulled-out cable was accomplished by compressed air trapped above the fluid contained in an accumulator which received the metered fluid from the cylinder. An ordinary gate valve in parallel with the hydraulic control valve per-



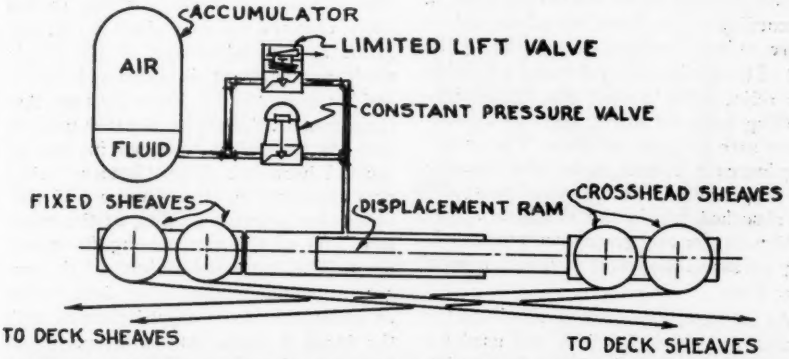

**EXP. TENSION-RAM TYPE ARRESTING UNIT ORIGINAL HYDRAULIC DESIGN 1928**

FIGURE 3.

mitted fluid to be forced back into the cylinder as the compressed air was allowed to expand. These developments resulted in considerable savings in weight and space, independence of ship's power and improved dynamics in the arrested landing provided by the inherent equal payout from each deck sheave.

The capacity of this unit was 8000 lbs. at 60 mph (950,000 ft. lbs.) The first shipboard installation was made in U.S.S. *Ranger* (CV-4) during her construction in 1934. Later, installations were made in *Lexington* and *Saratoga* (CV-2 and CV-3).

to save weight where full runout could not be allowed; aluminum parts were substituted for steel in some cases; welded tube cylinder construction was replaced by bored forgings; pressures were increased to 15,000 psi maximum; wire rope increased in diameter, etc. The more important developments occurred in reeving arrangements and methods of controlling hydraulic pressure. However, it is interesting to note that the basic design which was intended for ½" diameter, 22,000 lbs. breaking strength cable is still performing in a remarkable manner after being modified to take




**MK.4 DISPLACEMENT-RAM TYPE ARRESTING GEAR**
**U.S.S. HORNET (CV-8)**
**1941**

FIGURE 4.

The Mark 4 Mod O design (Figure 4), initiated to produce a unit of greater capacity, was generally similar to the Mark 3 except that the sheave arrangement was modified to use two shafts in tandem with five sheaves each instead of ten sheaves abreast on one shaft. Allowable cylinder pressure was increased from 5500 psi to 10,000 psi resulting in a unit capable of arresting 11,000 lb. aircraft.

Numerous modifications have been made in this basic design. However, they have been of little import as regards the progressive development of arresting gear. For example; special short ram (and cylinder) engines were built for use on the more forward arresting wires 11/16" diameter, 46,000 lbs. cable.

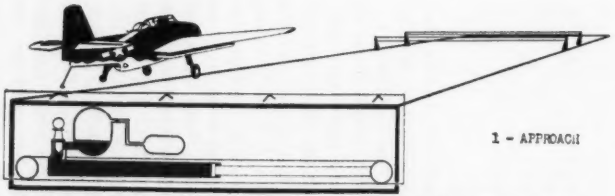
This is the gear with which we fought the major portion of the war, beginning with *Lexington* and *Saratoga*, *Ranger*, *Yorktown*, *Enterprise*, *Wasp* and *Hornet*, on through the early escort carriers of the *Bogue* and *Sangamon* classes and other conversions, *Independence* class light carriers, the old side wheelers *Wolverine* and *Sable* used for training on the Great Lakes, the first ten (10) *Essex* class medium carriers, *Casablanca* class, and the *Commencement Bay* class escort carriers. In addition, numerous installations were made at air stations in conjunction with an expeditionary type catapult for training pilots in catapulting and arrested landings.

The original control valve developed for Mark 4 arresting gear was a special spring loaded valve with an adjustable limited lift feature (controllable orifice). As this valve introduced peak cylinder pressures and peak loads in airplanes during arrestation, it was necessary to design a special air-loaded relief valve to limit such peak loads when airplane engaging speeds were high. This method of control operated satisfactorily but it was quite inefficient since the resulting pressure curve was in part a function of ram velocity, producing small area (little work done) for the maximum pressure (peak cable tension) developed. Accordingly, the functions of the valves were switched around—by reducing the lift of the limited lift valve and adjusting the relief valve to open at a desired controlling pressure rather than at a maximum safe cylinder pressure. The resulting increase in area under the pressure curve permitted much greater energy to be absorbed within the available stroke of the ram for any given maximum working pressure. *Constant pressure control was born.*

An attempt was made to eliminate the limited lift valve, but it is still used for three reasons: (1) it allows flow to begin at low pressure before the constant pressure valve opens, (2) permits pressure to descend gradually after the constant pressure valve has closed and (3) provides a simple means of opening the line to allow back flow from the accumulator and retrieve the engine.

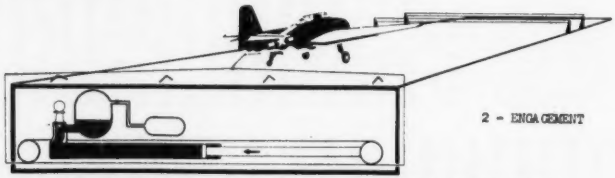
It became apparent early in the war that the capacity of Mark 4 arresting gear was inadequate since normal everyday operations required that the gear be used at nearly its maximum rated capacity. The main disadvantage to such operations was the high bearing loads and attendant large number of replacements necessary. Remember: bearings designed for 22,000# wire rope were being required to operate at 100% overload. Arresting gear of a greater capacity was needed for the *Essex* class carriers which would carry the most modern aircraft then on the drawing boards. Since arresting gear is equipment installed in ships, it is difficult and uneconomical to make basic changes in capacity after installation. Arresting gear capacity is, therefore, determined by analysis of trends in aircraft design and configuration, extrapolated many years into the future—if possible, for the expected life of the carrier. The Mark 5 arresting gear was designed on this basis for about 3 times the capacity of the Mark 4 gear.

The basic hydraulic design of the Mark 5 is the same as Mark 4, compression-displacement-ram type. Certain details, however, other than size, distinguish the Mark 5 design. For example; the tandem arrangement of sheave banks was replaced by a single group of ten (10) sheaves on one shaft as in the Mark 3. The advantage of the single shaft arrangement is discussed separately under "Wire Rope System Arrangements". All engine sheaves turn on anti-friction roller bearings in lieu of journal bearings. Also, "Insuroc" slippers replaced the top, bottom and side rollers for accurate guiding of the crosshead end of the ram during its movement. The increased weight of the engine required installation on deck in lieu of suspension from flight beams as with the Mark 4. Deck installation permitted mounting the air-liquid accumulator on top of the engine framework and elimination of long high pressure fluid lines. The major improvement in design is considered to be the control valve. The Mark 4 hydraulic control system, after introduction of constant pressure control, permitted only a meager degree of control for different energy levels which was obtained by adjusting the limit of lift on the orifice valve. An increase in lift only flattened the slope of the pressure curve for that portion of the cycle when the constant pressure valve was closed. Control in this manner made the engine inefficient since maximum hydraulic pressures were not reduced. Some means of altering the value of valve stem force tending to keep the constant pressure valve closed was necessary if the efficiency of constant pressure control was to be realized for all energy values. The Mark 5 hydraulic control valve provides this type of control by the use of an adjustable compound lever



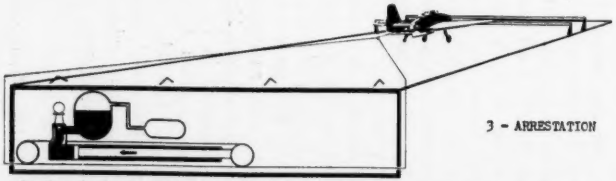
1 - APPROACH

With the deck pendants raised, the arresting engines at battery, barrier cables raised and the landing area clear the carrier is ready to receive aircraft. The approaching plane has its wheels down, flaps down and arresting hook down preparatory to landing.



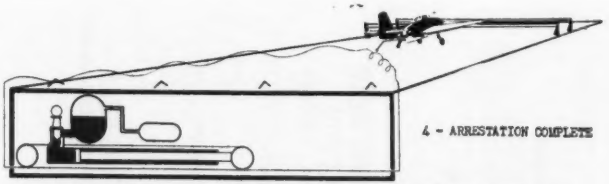
2 - ENGAGEMENT

The airplane hook engages the cross deck pendant pulling it forward, while the purchase cable is being pulled out from the deck edge sheaves.



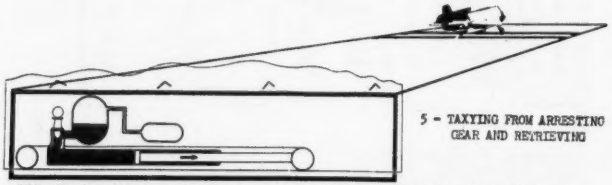
3 - ARRESTATION

As the airplane travels down the deck, purchase cable is pulled out causing the engine ram to be forced into the engine cylinder displacing hydraulic fluid under pressure through the control valve into the accumulator. This fluid metering process through the control valve restrains the pulling out of the purchase cable, and consequently the deck pendant.



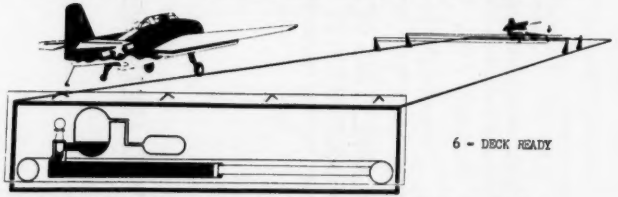
4 - ARRESTATION COMPLETE

The restraining action stops the motion of the airplane and upon stoppage the control valve automatically closes preventing the plane from being pulled aft along the deck by the deck pendant, an action known as "walkback". "Hookmen" run onto the flight deck and unhook the deck pendant from the airplane. Barriers are lowered, plane moves forward and pilot commences to fold wings.



5 - TAXYING FROM ARRESTING GEAR AND REYRIEVING

With the barrier down the plane moves over them and is "spotted" ahead of the barriers or taken down the elevator to the hangar deck. Simultaneously the deck pendant is retrieved to its ready position by allowing the fluid from the accumulator to flow back into the cylinder and force the ram to battery position.



6 - DECK READY

With the plane ahead of the barriers and the barriers raised, the landing area clear, and all arresting engines at battery, the carrier is ready to receive the next plane.

NORMAL LANDING SEQUENCE.

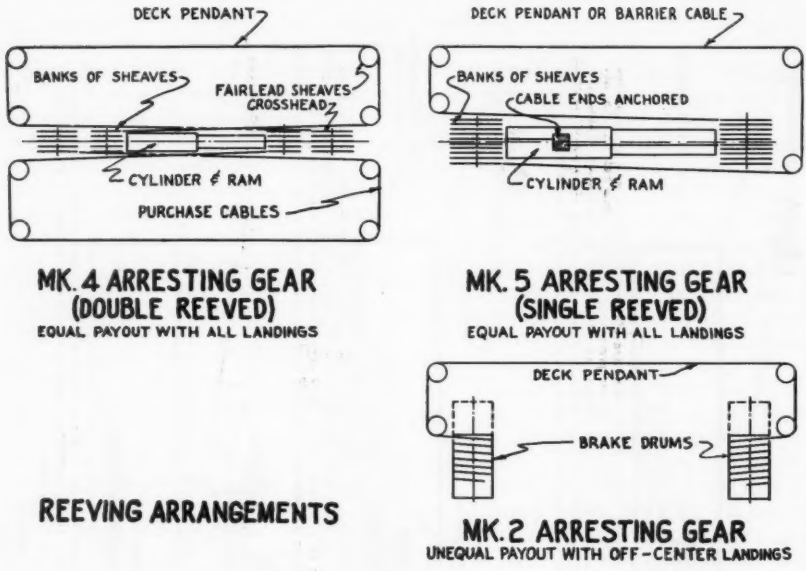



FIGURE 5.

system between the piston subjected to air pressure and the main valve stem. By mechanical movement of the fulcrum of one lever, the air load transmitted to the main valve stem is multiplied as much as four times. The function of a separate orifice (limited lift) valve is incorporated in the same valve body with provisions for mechanical opening to retrieve the engine.

The first Mark 5 arresting gear installation was made in U.S.S. *Bennington* (CV-20) in October 1944. The remaining vessels of this class received similar installations and some earlier vessels were converted, until at present Mark 5 arresting gear is installed in 16 *Essex* class CV's, 3 *Midway* class CVB's, and 2 *Saipan* class CVL's. The remaining 8 *Essex* class carriers may be converted in the future since their value as military weapons is increased upon the installation of Mark 5 or future improved arresting gear.

*Wire Rope System Arrangements* (Figure 5).

As stated before, all the basic essentials of arresting gear as used today were present when the first landing of an airplane was made aboard ship in 1911. Significant modifications to these essentials have been made over the years which are of interest.

On the *Pennsylvania* with the sandbag energy absorbers, the airplane was required to engage and pick up a series of individual athwartship cables since the effort exerted by any individual cable was very small. When the Mark 2 brakes were installed in the *Lexington*, a system of fore and aft cables was installed along with an athwartship cable system. The fore and aft cables, about 9″ apart, were secured to the deck at both ends and reached from the after knuckle of the flight deck to a point about 302 ft. forward. They were tensioned to 900 lbs. and supported about 15″ off the deck by four rows of "fiddle-bridges" 80 ft. apart. The fiddle-bridges could be raised or lowered at will and could be run down by airplane wheels without damage.

Each athwartship cable, which led to the brake unit at each side of the deck, was supported under the fore and aft cables by two "yielding elements" installed at the edge of the deck just outboard of the fore and aft wires. The seven athwartship cables were spaced 25 to 50 ft. apart but not all were raised at the same time. As engagement of only one cable was necessary to stop the airplane, four or five could be rigged and the remainder held in reserve in case of failure.

The fore and aft cables were intended to prevent the airplane from going over the side. Hooks installed on the airplane axle engaged the fore and aft cables and restrained side motion of the airplane. The trailing hook installed in the airplane engaged the athwartship cables which applied retarding loads.

Design of the deck sheaves which led the cables through the deck to the brake units provided for engagement of the cables during landings over the bow as an emergency measure.

The fore and aft system handicapped landing operations and caused minor damage to aircraft. As a result of tests conducted both ashore and aboard ship without the fore and aft wires, all such installations were removed after May 1929.

Development of the Mark 3 hydraulic unit which permitted the use of one unit for each arresting wire provided a saving in weight which was utilized for the installation of additional arresting wires, to improve the safety and reliability of arrested landings. War experience proved the validity of the premise and within practicable limits every foot of landing deck is considered when placement of arresting wires is planned. In the early days of the *Lexington* and *Saratoga* designers went one step further and installed arresting wires and barriers in the forward half of the flight deck to provide for landings over the bow with the ship backing down. Arresting wires were also installed in the barrier area as emergency wires. The same war experience showed that all this emergency gear was excess baggage since any battle damage which rendered the normal landing area incapable of repair and use could so seriously affect the vessel's ability to carry on that the emergency gear could not be used. Further, the vessel was vulnerable to attack for the 15 minutes required to reverse and while backing. Consequently, all bow arresting gear then installed was removed under authorization granted by the Vice-Chief of Naval Operations in July 1943.

There were no reeving problems to contend with when arresting gear was a collection of sand bags, but beginning with the *Langley* weight type it was necessary to provide a mechanical advantage to convert the tower height into a much greater runout distance on deck. Therefore, banks of sheaves were provided both at the top of the towers and on the tray which carried the weights in order that the distance the weights moved might be multiplied. A mechanical advantage of 8:1 was provided at each tower with each of the purchase cables being dead-ended at the top of the tower. Since only two pairs of towers were installed, additional arresting wires were provided by splicing in additional cables in parallel to each main cable of such length that the portion across the deck could be positioned an appreciable distance away. It can be seen that engagement of one of these cables by an airplane introduced slack into the parallel member creating a tangle of cables on deck.

To complicate the problem further, the weight trays did not move equal distances when aircraft landed to one side of the center span (off-center landing) since the airplane could pull more cable from the nearest tower and less from the tower further away. Under extreme off-center conditions, the result was a decrease in the effective capacity of the system. However, this feature did not become a serious problem with the small aircraft of that day.

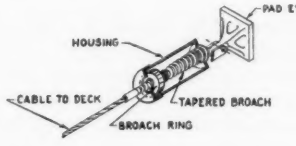
With the Mark 2 brakes, described earlier, each end of one arresting wire was wound around one brake drum—two drums or brakes for each wire. The

slack problem of the weight system was eliminated by increasing the number of absorbers but these brakes also payed out cable on demand with the same decrease in effective capacity during off-center landings. Unequal payout was inherent in the tension-ram type hydraulic unit as well. The Mark 3 hydraulic unit eliminated the difficulty of unequal payout since each engine was reeved with 2 lengths of wire rope and one end of each dead-ended on the unit frame while the remaining ends were led up on deck to provide the arresting wire. With this arrangement, equal payout was obtained under all landing conditions. It soon became apparent, however, that the dead ends could be led up on deck to provide an additional arresting wire without sacrificing other advantages gained. This arrangment was called "double reeving" and permitted either great savings in weight installed in the vessel or a greater number of arresting wires. Double reeving is the basic arrangement in use today. Single reeved engines are still used in cases where expansion joints or other ship structural design features do not permit passage of the additional purchase cable reeving.

Reeving of the Mark 5 engine is almost identical to Mark 4 with the exception of the position on the engine at which the cable ends emerge. The tandem sheave shaft arrangement of the Mark 4 requires that cables emerge from the sides of the engine at some fleet angle in both horizontal and vertical planes. The side by side or single shaft arrangement of the Mark 5 permits two leads to emerge at the horizontal center line and one lead at each side of the engine but within the confines of its frame-work. In fact, all leads emerge directly below the lead sheaves instead of to one side of the lead sheaves. In this manner all fleet is eliminated for the cables carrying the greatest load and which travel at the highest speed.

*Barrier Development.*

After the novelty of the first arrested landings had worn off and the practical and military potentialities of the aircraft carrier could be exploited, the first question to be answered was "how many aircraft can be used efficiently?" The studies of landing procedures, and spotting patterns for airplanes on the flight and hangar decks, would fill volumes. However, one fundamental was apparent. The flight deck must be divided into two sections—one for landings and one for parking and servicing operations—the latter where maintenance personnel could work in comparative safety in case of any failure or malfunction which would prevent a normal arrested landing. Some positive means of stopping such errant aircraft was mandatory because one wild airplane could wreck many more and do away with a good portion of the ship's crew in less time than it takes to tell it. Since damage to one airplane was not of particular concern for this emergency condition rather drastic measures could be taken as long as the pilot was not endangered. Wire ropes were suspended across the flight deck at sufficient height to engage on the airplane engine cowl and main wheel struts. As energy values at this time were low the barrier cables were secured to the deck with no working distance except stretch in the cables. The damage to the airplane and shock to the pilot can be visualized without further explanation, but for a time such barriers served a useful purpose.



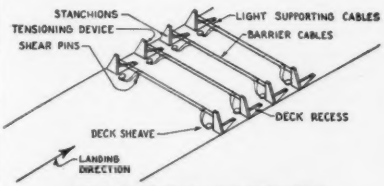

**BROACH DEVICE EARLY BARRIERS**

FIGURE 6.

The first energy absorber (Figure 6) designed specifically for barriers was a broach device. That is, a device which actually enlarged the hole in a metal ring. Greater or less energy could be absorbed by varying the thickness of the ring to be broached. The barrier cables were led below decks through hawse pipes and secured to the broach. No mechanical advantage was provided.

Consequently, the working distance was very short, and retarding loads were very high in comparison to the strength of airplane structure of that day. The broaches were used until after hydraulic arresting engines were standard equipment. At this time it was logical to use an arresting unit as energy absorber for the barriers identical to that for the arresting gear. There was one difference —the mechanical advantage was lowered to reduce the deck travel—since loads could be much higher than in a normal arrested landing. Barrier engines were reeved to a double six part purchase (6:1 mechanical advantage) instead of the double 10 part purchase (10:1 mechanical advantage) normally used for arresting engines.

Barriers were and are installed in multiple as an additional safety feature in case of failure or damage inflicted during a barrier crash. All barriers must be lowered to the deck after a landing to permit the airplane to taxi forward to the parking area or to clear the landing area for the succeeding airplane (Figure 7). Barrier cables must therefore be




MK 4 BARRIER ARRANGEMENT

FIGURE 7.

suspended from hinged supports or stanchions which can be raised or lowered at will by an operator in the flight deck walkway. This operation was accomplished manually in the early vessels such as *Lexington* and *Saratoga*. Consideration for the operators, however, led to the adoption of pneumatic operation controlled by a suitable air valve at the control station. The aircraft in service in the hey-day of these carriers included such amphibian aircraft as the old J2F as well as other varied configurations. Since the optimum height for barrier cable engagement varied with certain aircraft the early barrier stanchions were fitted with a series of stops such that their radial position could be controlled. Various barrier cable engagement heights were achieved in this manner. This latter feature was eliminated just prior to World War II when service carrier aircraft geometry became generally the same. Hence, we arrive at the standard Mark 4 barrier installation in the U.S.S. *Hornet* which was duplicated in all aircraft carriers built after her with the same arresting gear capacity.

All the basic features of the Mark 4 barriers were retained in the original Mark 5 design except one. In the Mark 4 design, the barrier engine purchase cable terminated on deck at a horizontal sheave mounted on the flight deck with both barrier cables being connected to one purchase cable by suitable turnbuckle fittings. In the Mark 5 Mod. 0 design a sheave was mounted in the stanchion itself and in the same plane such that the sheave was vertical with the stanchion in the raised position. The purchase cable emerged from a slot in the inboard edge of the stanchion and both barrier cables were secured to the purchase cable by pin type connections and a 3 cornered, bifurcated plate. The advantage expected from this feature was less severe damage to purchase cables on occasions when barrier cables became wrapped up in propellers and were subjected to a sharp up-haul against covers of horizontal sheaves—an occurrence which attended the introduction of 3 and 4 bladed props. It was intended that the Mark 5 stanchion would topple forward upon barrier engagement and permit greater freedom of cable movement in the slot provided in the stanchion.

When the design was tested ashore with catapulted dead loads it was found that wave action in the barrier cables caused the stanchions to be thrown to the deck with great force instead of being toppled as intended. This discovery upset arresting gear production tremendously since the *Essex* class carriers were being built to receive Mark 5 arrest-

ing gear, and there was no time left for redesign without delaying construction of badly needed vessels. To make the design serviceable, shock struts, constructed from airplane tail wheel oleos, were installed in the deck to absorb the energy of the falling stanchion thereby preventing its destruction. Seven CV's received the Mod. 0 design while manufacture of horizontal deck sheaves in accordance with Mark 4 principles was being accomplished. The modified design was installed as Mark 5, Mod. 1.

All but three Mark 5, Mod. 0 installations have been converted to Mark 5, Mod. 1. Also, the mechanical advantage has been changed from 6:1 to 8:1 to provide additional available runout and reduced barrier crash loads.

The simple double cable suspension type of barrier was admirably successful in stopping single propeller, tail wheel equipped airplanes without injury to the pilot and without appreciable damage when ample working distance was allowed. The advent of nose wheel equipped aircraft, first in the Grumman Tiger Cat—twin reciprocating engine, tricycle type, and thereafter a series of jet-propelled designs, made the suspended cable system obsolete for two reasons: (1) inadequate strength in the nose wheel installation to withstand the necessary retarding loads and (2) the practical certainty that twin propellers would pick up the barrier cables and wrap them up on the props and tear out the engines. All these conditions were determined under actual tests with real, but non-flyable airplanes.

Ever since the first F7F Tiger Cat was catapulted into a suspended cable in March 1945, a search has been conducted for a barrier which would stop both tailwheel and nosewheel equipped airplanes, as well as the suspended cable system stopped tailwheel airplanes.

Such a utopian barrier design has not yet put in an appearance. There is, however, a very unique barrier (or rather barrier adapter, since it is used in conjunction with the suspended cable or standard barrier) in current use for all nosewheel equipped aircraft except those with single propellers. It was conceived by the Naval Aircraft Factory, Naval Air Material Center, Philadelphia, primarily for the F7F Tiger Cat which was the main problem at that time. But it has proven satisfactory for jet aircraft as well. The original design involved the suspension of a non-load-carrying cable, of smaller diameter and strength than the barrier cable, between the barrier stanchions. The barrier cable (one instead of two) laid on the deck in a slack condition. Short lengths of cable spaced every few feet across the deck were secured to the suspended cable at one end and the barrier cable at the other. It was intended that the nosewheel strut engage the suspended cable and move it forward at airplane velocity. The resulting motion, simulating that of a static pendulum when its fulcrum is suddenly moved horizontally, would cause the barrier cable to be lifted and pulled into a bight which could be engaged by the main wheel struts. A prototype was built and tested with very *poor* results. A number of things were wrong, the most startling was the discovery that the suspended cable could be flung right into the cockpit, by at least one propeller, with lethal force. Reliability, in addition, was questionable. It remained for the project engineers to devise suitable substitute materials which would be of adequate strength but low in mass to eliminate the lethal potentialities, and devise methods to increase the reliability over a wide range of airplane speeds.

Part of the solution was found in the use of fabric parachute webbing for the suspended components which has a very high strength/mass ratio. Complete success was soon to follow and after rigid service test ashore—by the accumulation of some 200 barrier crashes, the majority of which were with the same airplane—the first shipboard installation was made in U.S.S. *Shangri-La* (CV-38) for evaluation of the F7F Tiger Cat as a carrier airplane in February 1946. Two barrier crashes occurred during the evaluation landings with only minor damage to aircraft.

Similar installations have been or are

being made in all active carriers to be ready for full-fledged operation of jet aircraft which are nosewheel type. Sufficient service experience has been accumulated to date to prove the reliability of this barrier adapter for jets as well as the twin propeller tricycle type of aircraft.

The search for a more versatile barrier design is continuing. Many, many proposals have been evaluated and discarded. Some are now reaching the prototype stage. Soon, we hope, a brand new principle will provide a multiple barrier installation capable of stopping any kind of airplane the aircraft designers can conceive.

If this article has held the reader's attention to this point, he may have gained from it an insight not only of the many problems which have been solved by a small group of men to make arresting gear what it is today, but also the tremendous part which an almost unknown art plays in the scheme of national defense. The value of the airplane in modern warfare is known to everyone, but few of us realize that the ability to deliver such a weapon at the enemies' front door depends on arresting gear. If necessary, even the sub-sonic aircraft that are flying today can take off from carrier decks under their own power *but* they cannot get back aboard without assistance.

With the importance of arresting gear established, it is not hard to visualize the responsibilities facing designers in the provision of future equipment which must be satisfactory for any aircraft the manufacturers can dream up for some years to come. The arresting gear requirements for present jet fighter aircraft involve problems of great magnitude which must be solved very quickly if arresting gear is to anticipate the demands of future aircraft.

# OF POSSIBLE INTEREST TO MEMBERS.

## VETERANS' DEADLINES

UNTIL MARCH 31, 1949: Veterans have preference in renting or purchasing new homes and apartments, under Federal rent control law.

UNTIL APRIL 20, 1949: Eligible next-of-kin may apply for gratuitous insurance benefits of those veterans who were entitled to gratuitous NSLI; those who died in the line of duty after October 8, 1940, and prior to April 20, 1942.

UNTIL JUNE 30, 1949: Applications may be made by amputees for free automobiles not in excess of $1,600.

UNTIL JULY 25, 1952: Payments of Readjustment Allowances under GI Bill end.

UNTIL JULY 25, 1952: Competition for all Federal Civil Service positions of guards, elevator operators, messengers and custodians, is restricted to veterans only, in cases where sufficient numbers of qualified veterans are available.

# LINES—A discussion of some of the factors affecting the choice of a transmission line type for a specific application in Navy electronics.

BY LIEUTENANT COMMANDER CHARLES W. HARRISON, JR., U.S.N.

THE AUTHOR.

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In 1945 Lieut. Comdr. Harrison was naval liaison officer at the Evans Signal Laboratory, Belmar, New Jersey. In early 1946 he matriculated at Harvard to continue work on the degree of Doctor of Science in Engineering. Later in that year he reported for duty at the Philadelphia Naval Shipyard where he served as Acting Electronics Officer for approximately a year. Other assignments included duty as Electronics Planning Officer and Assistant for Shore Electronics.

Lieut. Comdr. Harrison is presently attached to the Electronics Design Division, Bureau of Ships. He is a member of the Harvard Chapter of Sigma Xi; the American Physical Society; the Institute of Radio Engineers; and the Audio Engineering Society. He is registered as an electrical and radio engineer in Virginia.

*Introduction.*

Knight's Modern Seamanship (10th Edition) defines a "line" as "a general term for light rope." In the modern Navy this definition must be regarded as incomplete, for extensive use is made of transmission lines for various purposes in Navy electronics. The present paper is confined to a discussion of the relative merits of several classes of lines, including wave guides, for various military applications. Comments are restricted to electrically long lines; i. e., lines comparable to a wavelength in overall length. (The voltage and current distribution along an electrically long line may or may not be uniform, depending upon the termination.) Falling into this category, for example, are video cables employed for the interconnection of electronics equipment aboard ship, and transmission lines used for conveying the radio frequency power output of a transmitter into a directional antenna at a shore radio station.

Some of the more important applications of transmission lines and line sections, in addition to their obvious function of transferring power from one location to another, include use as impedance transformers with low losses, high Q tank circuits in oscillators and tuning circuits in microwave receivers, filters and harmonic suppressors, as well as pulse shaping networks or rectangular wave generators. In many microwave circuits line sections are employed as insulators! Only recently a "distributed" video

amplifier was built which employs transmission line sections and has a band pass width of several hundred megacycles.

In the Navy considerable use is made of the two-wire line with air dielectric, coaxial cables, and wave guides. The four-wire line with air dielectric (having the diagonal wires connected in parallel) is probably not employed as much as its excellent characteristics warrant. The two coaxial cables in general use are the gas (or air) dielectric and solid dielectric types.

The writer does not propose to describe in detail the physical appearance of any transmission line mentioned above nor make any statements concerning line constructional practice. Suffice to say that open wire lines (whether of the two or four-wire type) are supported at intervals by insulators secured to poles. Additionally, the insulators insure that the spacing between wires is uniform. One coaxial cable type consists of a cylindrical copper tube within which is supported a solid copper wire of cylindrical shape. Small ceramic discs equally spaced along the line act as insulators and supports for the center conductor. Ordinarily such lines are dehydrated and then an inert gas, such as nitrogen, is kept in the line (under pressure) to prevent moisture from entering. Trunks, as used aboard ship, may be regarded as a form of coaxial cable. Usually trunks are not moisture-proof. The solid dielectric type of coaxial line consists basically of a woven stranded outer conducting sheath and a center conductor (either of solid or of stranded copper wire) immersed in a low-loss solid dielectric. Wave guides are relatively thin-wall copper tubes of either rectangular or circular cross-section, depending on application.

*Factors Effecting the Choice of a Line Type for a Specific Application Aboard Ship.*

One would be extremely naive to suppose that extensive use of open-wire lines could be made aboard ship. The vast number of electrical interconnections required between compartments of a modern naval vessel would preclude the use of appropriately spaced open-wire lines, for not only would space limitations prevent this but it would be extremely difficult to achieve compartment integrity (from the point of view of water-tightness); the shock hazard would be ever present; switching arrangements are cumbersome; and it would be difficult if not impossible to obtain satisfactory electronics equipment operation due to coupled circuit effects existing between the maze of wires. Further, at very high frequencies (when the spacing between wires of the line becomes comparable to the wavelength) an open-wire line may be regarded as a special form of antenna array. Under such circumstances much of the power supplied to the line may be radiated; little or no power being delivered to the load. The objective of the line is thus defeated. The use of either a coaxial cable or waveguide is indicated.

The longest wavelength or dominant mode that can be transmitted by a hollow conductor of rectangular cross-section having sides a and b (inside dimensions) ($a > b$) is determined from inequalities $a > \frac{\lambda_{TEM}}{2}$ and $b < \frac{\lambda_{TEM}}{2}$ ($TE_{1,0}$ mode). Here $\lambda_{TEM}$ is the wavelength taken in empty space ($\lambda_{TEM}$ in meters equals $3 \times 10^8$ meters per second divided by the frequency in cycles per second). There is no lower limit for the value of b, exclusive of zero. It can be made as small as desired, but attenuation increases with decreasing values of the dimension b.

The longest wavelength or dominant mode that can be transmitted by a hollow conductor of circular cross-section having inside radius b is determined from the inequality $b > \frac{\lambda_{TEM}}{3.41}$ ($TE_{1,1}$ mode).

Suppose it is desired to convey radio frequency power having a frequency of 25 megacycles per second from a radio transmitter aboard ship to a topside antenna. For this case, $\lambda_{TEM}$ equals 25 meters or 82 feet. If a waveguide of rectangular cross-section is employed the dimension a will have to equal or exceed 41 feet! If a waveguide of circular cross-section is suggested for this application, its radius will have to be in excess of 24 feet! As a matter of interest, no waveguides of circular cross-

section are in general use in the Navy, exclusive of their use as waveguide fittings for rotating joints, etc. No rectangular waveguides are employed aboard ship for frequencies below twelve hundred megacycles per second.

From the above discussion it is clear that waveguides have no application at communication frequencies. Accordingly, the use of a coaxial cable is indicated. The question is whether the gas-filled line or the solid dielectric cable should be employed. Unfortunately, the solid dielectric coaxial cable has higher inherent loss (attenuation) than the gas-filled line. But the gas-filled line suffers from the disadvantage that it must be maintained gas tight. This is not easy to accomplish on a flexing ship. If all lines used aboard ship for electronics applications were gas-filled, rupture of the lines in battle would certainly be a personnel hazard.

It appears, therefore, that in spite of its high losses the solid dielectric coaxial cable is the optimum line for shipboard applications (exclusive of high tension applications, involving long and medium wave transmitters, where trunks are necessary) provided the frequencies involved are not too high. Such a line is of little or no use in microwave applications, as it is entirely possible for a relatively short line to absorb the entire power output of a transmitter, thus delivering no power to the antenna. Research is being conducted toward the development of a dielectric material which is not only low-loss, but impervious to moisture. It is worth mentioning that the absorption of moisture by the line is accompanied by severe attenuation of the signal.

The solid dielectric cable is easy to "pull" into position aboard ship and requires essentially no maintenance provided the cable terminations (or fittings) are properly installed by shipyard mechanics. Considerable forethought must be exercised in preparing installation drawings for such cables, as many electrical failures are directly attributable to the cables' proximity to hot stacks, fire rooms, etc. It is hoped that future development will evolve a good low-loss, moisture-impervious, high-temperature-resistant cable for general shipboard use.

Rectangular waveguides are used exclusively for long hauls where microwave frequencies are involved. Happily, the loss of power in the guide is small if it is properly designed and installed.

*Factors Effecting the Choice of a Line Type for a Specific Application Ashore.*

Open wire transmission lines have a very important application at shore radio stations.

The two-wire line is well adapted for connecting high power transmitters (operating at communication frequencies) to balanced antennas, such as rhombics installed on open territory. (By definition an antenna is balanced if it is symmetrical with respect to its input terminals from the point of view of the geometrical arrangement of the conductors forming the two halves of the radiating system, and with respect to nearby objects, such as the earth.)

The fact that a two-wire line is generally satisfactory for use at transmitting stations ashore (except where climatic conditions preclude their use) is not prima-facia evidence that such lines are useful, for example, for connecting a directive receiving antenna to a receiver. Prime requisites for an effective line for use in receiving applications are small radiation resistance (or effective length) and symmetrical installation. Although the reciprocity theorem shows that the transmitting and receiving qualities of a given line are identical, it is erroneous to conclude that a negligible contribution to the radiated power when trans-

mitting implies a negligible pickup in reception. There are fields acting on the line set up by numerous moving charges in other antennas. The interfering fields may be of sufficient amplitude to cause appreciable undesired voltage development across the input circuit of the receiver. The four wire transmission line is particularly well suited for connecting directive receiving antennas to receivers at communication frequencies. The coaxial cable may be adapted for transferring power from the output circuit of a transmitter to a radiator (whether symmetrical or not) provided the outer surface of the coaxial cable is in contact with the surface of a highly conducting earth along its full length. Preferably the cable should be buried in the ground. If a coaxial cable is employed (having a solid copper sheath) to connect two balanced impedances, and the outer surface of the cable is not properly grounded, the line often acts like a three-conductor transmission line. (It is assumed that the frequency of the voltage applied to the sending end of the line is sufficiently high so that skin effect is an important consideration. Under these circumstances it is possible to distinguish currents flowing on the inside surface of the tube from those flowing on the outside surface of the same tube.)

As stated earlier, one important advantage of the gas-filled coaxial line over the solid dielectric cable is that its dielectric losses are lower. But this advantage is perhaps partly offset by the fact that the solid dielectric cable is more easily installed at shore radio stations. It is extremely difficult to install and maintain a gas-tight line, say 1,000 feet in length. Invariably gas leaks occur. The writer understands that an important communications company had such a difficult experience in this regard that all coaxial cables were dug up and eventually sold to a South American country in the interest of "good will relations"! The location of gas leaks along a buried coaxial cable is not a simple matter. An engineer of the concern referred to above suggested one procedure for determining the location of leaks. Bird scent (which is non-corrosive) is acquired and pumped into the line. The best available hunting dogs are then employed for finding the leaks!

The employment of flexible solid dielectric cables is advantageous, for example, in mountainous terrain and in tropical swamps, provided its relatively-high copper and dielectric losses can be tolerated. This presupposes the cable, when used for transmitting, will withstand the voltages encountered when severe mismatch conditions obtain. Radiation, or conversely pickup, are negligible if the cable is in proximity to a highly conducting earth regardless of whether balanced or unbalanced load impedances are involved. (The earth is not a good conductor at frequencies higher than those presently employed for a-m broadcasting.)

*Comments on Lines.*

Some engineers assume a priori that because the conventional analytical treatment of transmission lines fails to reveal certain characteristics that these are either nonexistent or at most are of no special import. For example, it is taken for granted that a "perfectly balanced open-wire line will not pick up radiations" because the analysis of transmission lines based on conventional circuit theory includes no radiation phenomenon. Such an assumption is equivalent to denying the possibility of radiation from a two-element or multi-element antenna array! Likewise, such phenomenon as proximity effect (the axial asymmetrical orientation of charge about each conductor forming a line) and skin effect (except as it is reflected in the value of the attenuation factor) are ignored. (Skin effect is of importance in determining, for example, when a coaxial cable behaves like a three-conductor transmission line.) Only by a fundamental analysis of the line, based directly on Maxwell's equations are all of the factors influencing line operation determined.

The solution of the problem of the two-wire line as based on conventional circuit theory provides the immediate solution for the coaxial cable and for the multiple-wire, single-phase lines with regard to voltage and current distribution along the lines. Ordinarily, one derives line theory on the basis of the quasi-stationary state. On the other hand the waveguide problem is one requiring a direct application of electromagnetic theory.

In general the currents flowing along a transmission line (for example, a two-wire line) may be of two kinds. "Transmission line currents" are defined as currents of equal magnitude but of opposite phase at a given point along the line. Currents of this type flow along lines when the lines are balanced with respect to near-by objects, and provided balanced impedances are being connected together. On the other hand, if a balanced transmission line is being used to drive an asymmetrical impedance, such as a dipole driven at some point other than the center, or to drive elements of a collinear array, "antenna currents" will flow along the line, and the line becomes part of the radiating circuit. This phenomenon might be better understood by reference to Figure 1. Here an antenna is center-driven by a two-wire line, but the line is asymmetrically arranged with respect to the antenna. The lower half of the antenna reacts on the line more than the upper half, and unbalanced line currents obtain. Suppose the current in wire (1) (where the line is sectionalized) is $I_1$, and in wire (2) it is $I_2$. Now $I_1 \neq I_2$. Define the transmission line current where the line is sectionalized $I_{1L}$ and $I_{2L}$. The antenna currents are $I_{1L}$ and $I_{2L}$. Then

$$I_1 = I_{1L} + I_{1A}$$
$$I_2 = I_{2L} + I_{2A} = -I_{1L} + I_{1A}$$

The currents $1_{1A}$ and $I_{2A}$ are of equal amplitude and flow in the same direction. The two conductors of the line act like a single conductor antenna carrying a current $2I_{1A}$. If a coaxial line is substituted for the open wire line, current may flow on the outside surface of the cable giving rise to substantial radiation.

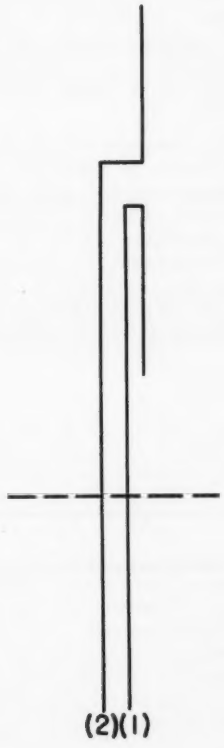


FIGURE 1.—Transmission line asymmetrically arranged with respect to the input terminals of a symmetrical center-driven antenna.

The ohmic loss along either the two or four wire lines is small if the line is short regardless of how the line is terminated. Physically long lines should be operated in a non-resonant condition by terminating them in their characteristic impedance. Conversely, standing waves, from the point of view of loss, may be tolerated along physically short lines. Radiation loss (as well as ohmic loss) is a function of frequency. Radiation from two wire transmission lines, when used in transmitting applications, may be ignored provided the line carries what might be termed a true transmission line current. (Symmetrically arranged wires do not necessarily constitute a balanced line. "Balanced" open wire lines feeding asymmetrical loads act like antennas, as explained earlier.) Radiation or conversely pickup of properly installed four wire lines is negligible.

*Calculation of Radiation Resistance of Two and Four Wire Transmission Lines.*

The determination of the radiation resistance of an open wire transmission line is of considerable importance for by the reciprocation theorem, the radiation resistance is a direct measure of the merit of the line as a receiving antenna.

Under certain circumstances antenna theory may be applied directly to transmission lines to predict the power radiated by them. In particular, this is true for a line any integral number of half-waves in length, driven at one end and open-circuited at the other end, i. e., for 100% reflection. Happily, the current distribution is essentially sinusoidally distributed, and this may be assumed at the outset with negligible error. (In antenna theory one can not assume the current distribution to be of a certain form. One important phase of the antenna problem is to determine the current, subject to existing boundary conditions.)

The objection might be raised that a transmission line when employed for transferring power from one location to another, as distinguished from its use as an impedance transformer, is never operated under conditions of 100% reflection. This is a valid objection, but a rather intricate analysis of the problem reveals that under several specific conditions of loading, including non-resonant operation, the radiation resistance is never greater than that computed for the case of 100% reflection. This establishes a criterion for comparing in this respect, the performances of several different open-wire lines.

An anomalous result will be obtained unless the analysis for radiation from a non-resonant line is based on one of the following premises:

1—The line system (including terminations) form a closed system of conductors carrying a progressive current wave.

2—The line system does not form a closed system, but a progressive current wave exists on conductors terminated by charges, such that the condition of electrical continuity at the ends of the wires obtains.

Thus any determination of radiation resistance, purporting to apply to non-resonant transmission lines, must be based on premise (1) or (2).

Consider a two wire transmission line an integral number of half-waves in length driven at one end and open circuited at the other end. Refer to Figure 2A. The current distribution along the wires is sinusoidally distributed to a high degree of approximation, implying a vanishingly small current at each end of the line. The line as described may be replaced by an

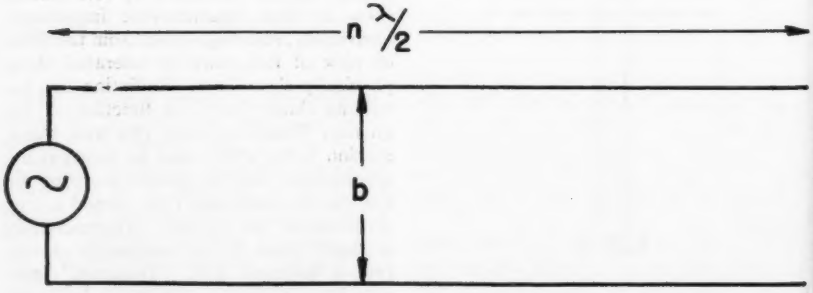


FIGURE 2A.—End driven two wire transmission line a multiple of a half-wave in length. The line is open circuited at the receiving end. The current distribution along the line is essentially sinusoidally distributed.

antenna array, as shown in Figure 2B, composed of two symmetrical center-driven radiators. The two driving generators are identical and maintain the flow of currents of equal amplitude, but of opposite phase along the wires. The current distribution along the wires of the two element close-spaced array (Figure 2B) is not significantly different from the current distribution along the transmission line (Figure 2A). The radiation resistance of the line is the same as the radiation resistance of the array, and the latter may be easily determined in terms of the available expressions for mutual and self-impedance for coupled antennas. It is to be observed that the generators are located at a current node (point of high impedance). Expressions for mutual and self-impedance, based on the assumption of a sinusoidal current distribution are meaningless when referred to a current node, accordingly, such formulas are referred to a maximum current existing along the radiator. The first current maximum exists one quarter wavelength from the free ends of either antenna.

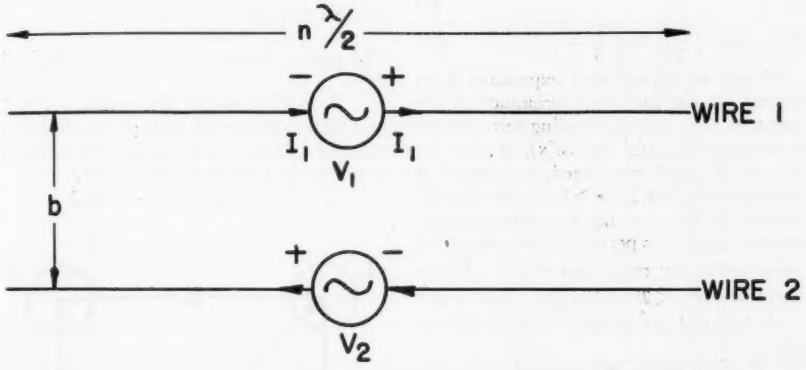


FIGURE 2B.—Transmission line (or two element antenna array) driven so that the current distribution along the wires closely approximates the current distribution along the two wire line shown in Figure 2A.

Referring now to Figure 2B, the relation between the voltages $V_1$ and $V_2$ and the currents $I_1$ and $I_2$ are given by

$$V_1 = I_1Z_{11} + I_2Z_{12}$$
$$V_2 = I_2Z_{22} + I_1Z_{21}$$

Here $V_1$ and $V_2$ are the voltages applied at the centers of wires 1 and 2 respectively. $I_1$ and $I_2$ are the currents flowing into or out of the input terminals located at the centers of wires 1 and 2, respectively. $Z_{11}$ is the self-impedance of antenna 1; $Z_{22}$ is the self-impedance of antenna 2; and $Z_{12} = Z_{21}$ is the mutual impedance of wires 1 and 2.

In the present case $V_1 = -V_2$, $I_1 = -I_2$, $Z_{11} = Z_{22}$ and $Z_{12} = Z_{21}$. Accordingly,

$$V_1 = I_1Z_{11} - I_1Z_{12} = I_1(Z_{11} - Z_{12}).$$

The input impedance of antenna 1 is

$$Z_1 = V_1/I_1 = Z_{11} - Z_{12}.$$

It is evident that the input impedance of antenna 2 is precisely the same as the input impedance of antenna 1 because of symmetry.

The power radiated by wire 1 is

$$P_1 = I_1^2(R_{11} - R_{12})$$

where $R_{11}$ is the resistive component of the self-impedance, and $R_{12}$ is the resistive component of the mutual impedance.

The power radiated by wire 2 is the same as the power radiated by wire 1. Hence the total power radiated is

$$P_0 = 2I_1^2(R_{11} - R_{12})$$

and the radiation resistance referred to the line current at a point of current maximum is

$$R_{\max\ (2\ \text{wire line})} = P_0/I_1^2 = 2(R_{11} - R_{12}) \quad (1)$$

From elementary theory of the multiple half-wave antenna

$$R_{11} = 30\left\{0.5772 + \log_e 4\beta h - \text{Ci}\, 4\beta h\right\} \quad (2)$$

$$R_{12} = 30\left\{2\,\text{Ci}\,\beta_b - \text{Ci}\,\beta\left(\sqrt{(2h)^2+b^2}+2h\right) - \text{Ci}\,\beta\left(\sqrt{(2h)^2+b^2}-2h\right)\right\} \quad (3)$$

Here $\beta = 2\pi/\lambda$, $\lambda$ is the wavelength, h is the antenna half-length, (i. e., the overall length of the transmission line is 2h) and b is the distance between wires, measured center-to-center.

For small values of x, $(-17 \leqq x \leqq 17)$ the following series are good approximations for Ci(x) and Si(x):

$$\text{Ci}(x) = 0.5772 + \log_e x - \frac{1}{2}\frac{x^2}{2!} + \frac{1}{4}\frac{x^4}{4!} - \ldots \quad (4)$$

$$\text{Si}(x) = x - \frac{1}{3}\frac{x^3}{3!} + \frac{1}{5}\frac{x^5}{5} - \ldots\ldots\ldots\ldots \quad (5)$$

If only an approximate expression is required for the radiation resistance of the two wire line, only the leading terms of the expansions for Ci(x) and Si(x), as given by (4) and (5) need be retained, and when it is remembered that $2h >> b$, a great simplification in the resulting equations may be secured, since it is permissible to neglect all terms involving small quantities. {Note that when $b << 2h$, $\beta\left(\sqrt{(2h)^2+b^2}+2h\right) \doteq 4\beta h$ and $\beta\left(\sqrt{(2h)^2+b^2}-2h\right) \doteq \frac{\beta b^2}{4h}$.} Care must be exercised, however, to insure that a sufficient number of terms of higher order than the second power are retained, for otherwise when forming an expression for the radiation resistance of a particular multiple wire line, the result may vanish.

When this procedure is followed the radiation resistance for a two wire line is found to be

$$R_{\text{max (2 wire line)}} = 120\pi^2\left(\frac{b}{\lambda}\right)^2 \text{ ohms} \ldots\ldots (6)$$

The radiation resistance of a four wire single phase transmission line may be determined as follows: Refer to Figure 3. Wires 1 and 3 are connected in parallel and wires 2 and 4 are connected in parallel, i. e., conductors of the same polarity lie diagonally opposite each other. (If wire 1 carries a current in phase with the current in wire 2, and these currents flow in the opposite direction with regard to the currents flowing in wires 3 and 4, little advantage is achieved over the use of a simple 2 wire line and this case will not be considered here.) The four wire transmission line is now replaced by a four element antenna array, each element of which is center-driven and carries currents of the same magnitude, but of the requisite phase.

Employing the previously used notation one has

$$V_1 = I_1Z_{11} + I_2Z_{12} + I_3Z_{13} + I_4Z_{14}$$

But

$$I_1 = I_3 = -I_2 = -I_4$$

and

$$Z_{12} = Z_{14}$$

Accordingly,

$$V_1 = I_1(Z_{11} + Z_{13} - 2Z_{12})$$

The input impedance is

$$Z_1 = V_1/I_1 = Z_{11} + Z_{13} - 2Z_{12}$$

or the input resistance is

$$R_1 = R_{11} + R_{13} - 2R_{12}$$

The power radiated by wire 1 is obtained from

$$P_1 = I_1^2 R_1 = I_1^2 (R_{11} + R_{13} - 2R_{12})$$

and by symmetry the total power radiated, referred to the maximum current existing in one wire of the four-wire line, is

$$P_0 = 4I_1^2 (R_{11} + R_{13} - 2R_{12})$$

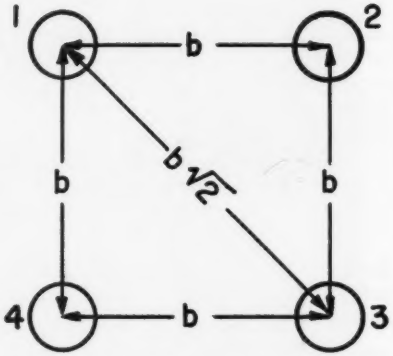


FIGURE 3.—Four wire transmission line. The wires are oriented at the corners of a square. Diagonal wires are connected in parallel.

But the *line current* $I_0$ is twice the current existing in one wire, i. e., $I_0 = 2I_1$, $I_0^2 = 4I_1^2$. Therefore, the power radiated, referred to the total line current, is

$$P_0 = I_0^2 (R_{11} + R_{13} - 2R_{12})$$

and the radiation resistance, referred to the maximum value of the total line current, is given by

$$P_0/I_0^2 = R_{11} + R_{13} - 2R_{12} \ldots\ldots (7)$$

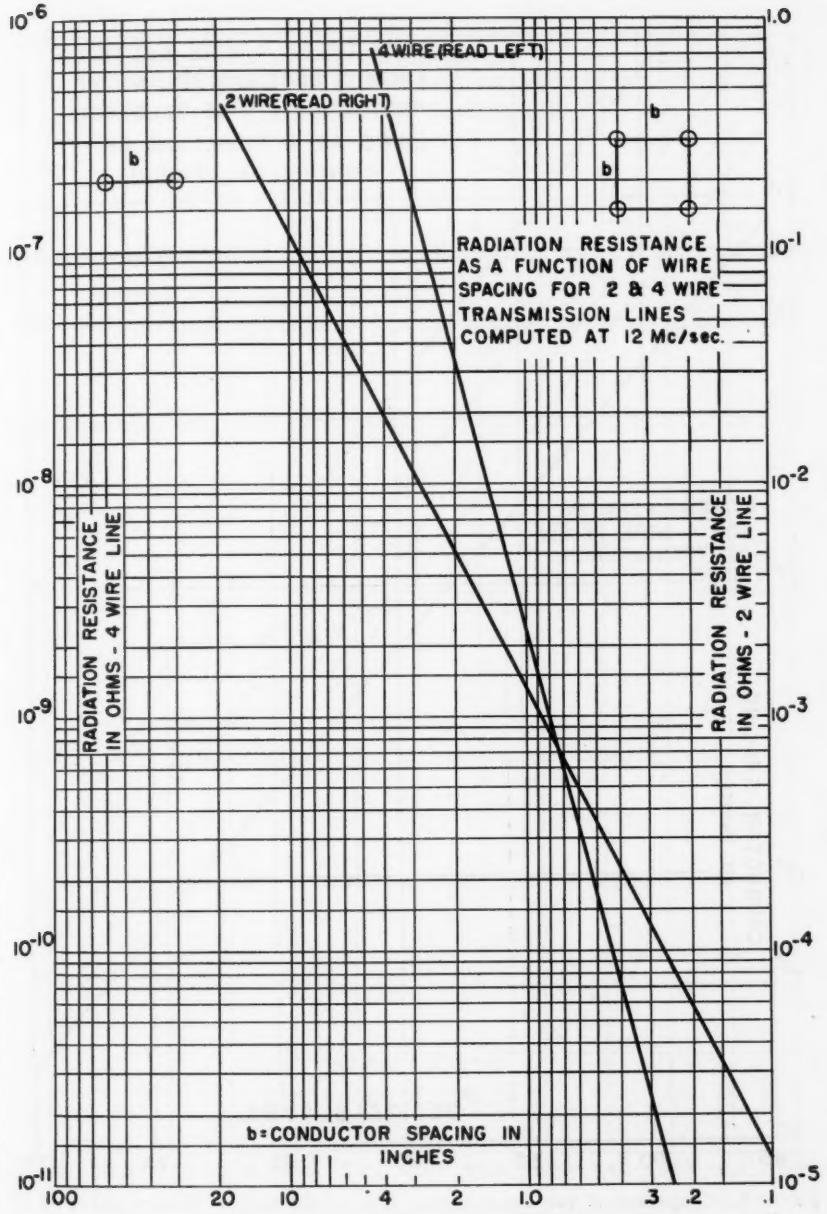


FIGURE 4.—Radiation resistance (referred to the current maximum) of two and four wire transmission lines as a function of wire spacing at 12 Mc/sec.

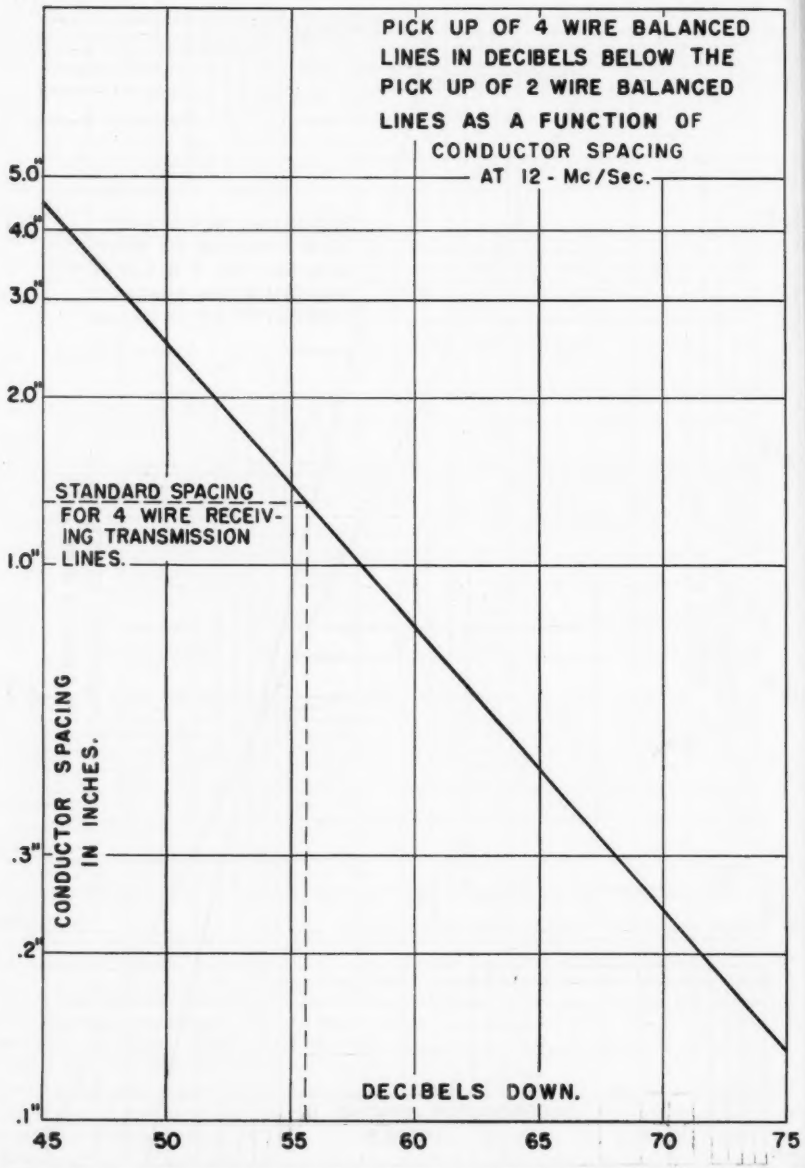


FIGURE 5.—Comparison of two and four wire lines on the basis of relative pickup at 12 Mc/sec.

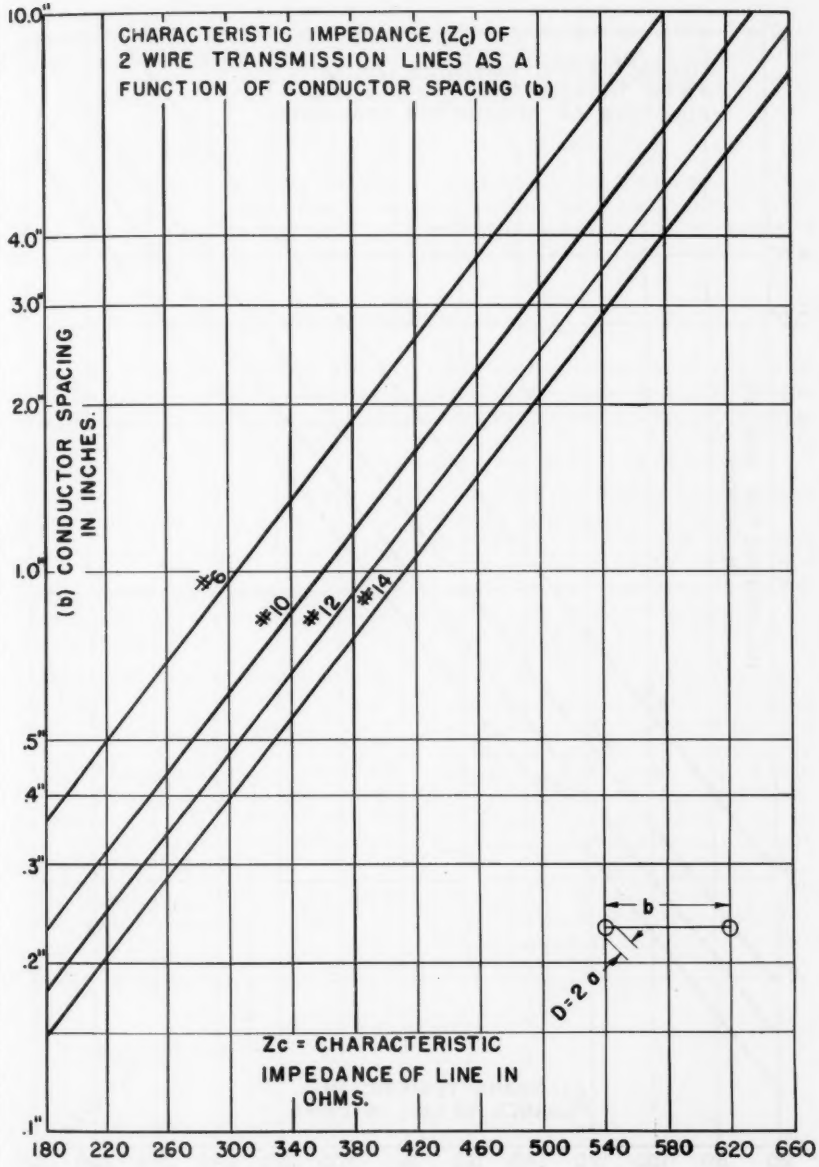


FIGURE 6.—Characteristic impedance of two wire transmission lines.

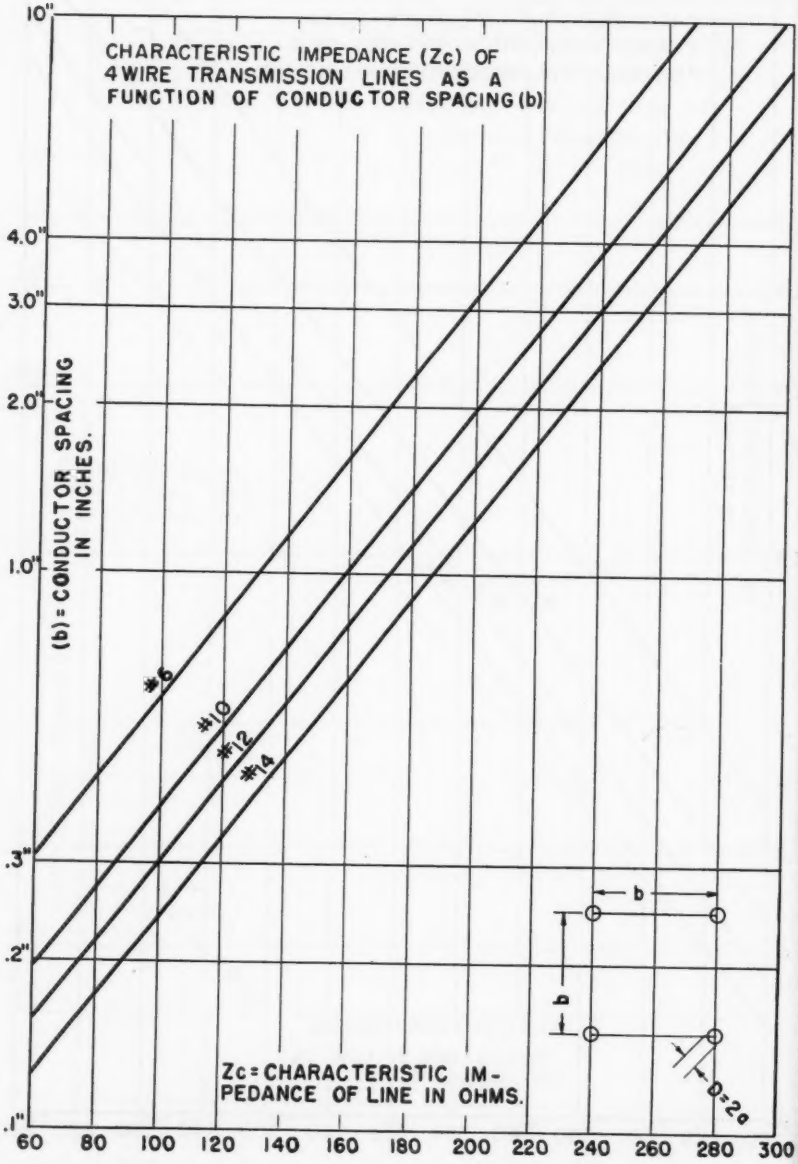


FIGURE 7.—Characteristic impedance of four wire transmission lines.

When the calculations are completed, subject to similar approximative processes employed in the case of the two wire line, one finds that the radiation resistance of a four wire transmission line is

$$R_{max\ (4\ wire\ line)} = 20\pi^4 \left(\frac{b}{\lambda}\right)^4 \text{ ohms.}$$

It is apparent, from a consideration of the equations for the radiation resistance of two and four wire transmission lines that the power radiated by a four wire line is extremely small compared to that radiated by a two wire line, for the same frequency and comparable spacing. By use of the reciprocity theorem, the signal pickup of a four wire line is extremely small compared to the pickup of a two wire line. The pickup of the four wire line, in decibels below the pickup of a two wire line of the same spacing is

$$db = 10 \log_{10}\left(\frac{6\lambda^2}{\pi^2 b^2}\right) \quad \ldots\ldots\ldots\ldots (8)$$

Two and four wire transmission lines are compared on the basis of their radiation resistances and their relative pickups in Figures (4) and (5).

*Characteristic Impedance of Conventional Transmission Lines.*

The behavior of a transmission line depends in part on the characteristic impedance of the line. This impedance is a pure resistance if the line is presumed to be lossless, i. e., $Z_c = R_c$.

For a two wire transmission line the characteristic impedance $Z_c$ is

$$Z_c = 276 \log_{10}\frac{b}{a} \text{ ohms} \quad \ldots\ldots\ldots\ldots (9)$$

Here b is the distance between wires, measured center to center. a is the radius of the wires employed.

The characteristic impedance of a four wire line is

$$Z_c = \left(138 \log_{10}\frac{b}{a}\right) - 20.78 \text{ ohms} \quad . (10)$$

Thus the characteristic impedance is 20.78 ohms less than one half the characteristic impedance of a two wire line having the same wire spacing. (The above formulas are valid only when $b \geqq 10\,a$).

The characteristic impedance of a concentric cable is

$$Z_c = \frac{138}{\sqrt{\epsilon_r}} \log_{10}\frac{b}{a} \text{ ohms} \quad \ldots\ldots\ldots (11)$$

where b is the distance between the center of the inside conductor to the inside surface of the outer tube, a is the radius of the center conductor and $\epsilon_r$ is the relative dielectric constant, 1 for air.

The characteristic resistances of two and four wire lines are shown in Figures (6) and (7) respectively. Notice that the range of coverage is different for the two lines. The coaxial cable further extends the range of characteristic resistance downward. The importance of the four wire line as an impedance transformer should not be ignored. For example, suppose it is desired to match a resonant center-driven dipole having an input resistance of 70 ohms to a two wire transmission line having a characteristic resistance of 600 ohms. A quarter-wave transformer may be employed for this purpose. The characteristic resistance of the quarter-wave transformer is given by

$$R_c = \sqrt{70 \times 600} \doteq 205 \text{ ohms.}$$

Reference to Figure (7) reveals that this value of $R_c$ is well within the range of four wire transmission lines having various design parameters.

# NOTICE—Activities of the Joint Research Committee on Boiler Feedwater Studies for 1948.

At the annual meeting in 1947 new officers of the Joint Research Committee were elected. These officers are:

*Chairman*
Dr. W. C. Schroeder, Chief,
Office of Synthetic Liquid Fuels,
Bureau of Mines,
U. S. Department of the Interior,
Washington 25, D. C.

*1st Vice Chairman*
Dr. S. F. Whirl,
Chief Chemist,
Duquesne Light Company,
435 Sixth Avenue,
Pittsburgh 19, Pennsylvania.

*2nd Vice Chairman*
Mr. R. W. Seniff,
Engineer of Tests,
Baltimore & Ohio R.R.,
Pratt & Arlington Avenue,
Baltimore, Maryland.

*Secretary*
Dr. T. H. Daugherty,
Research Chemist,
Hall Laboratories, Inc.,
323 Fourth Avenue,
Pittsburgh, Pennsylvania.

During the year there were no changes in the membership of the Executive Committee with the exception that Mr. E. H. Tenney resigned and Dr. S. F. Whirl was appointed in his place to represent the Edison Electric Institute. During the year the Engineering Experiment Station, Department of the Navy resumed its participation in the work of the Joint Research Committee, and Mr. R. C. Adams has been appointed as their representative on the Executive Committee.

The activities of the committee were generally concerned with the sponsorship of programs and papers at various society meetings. In May a program was sponsored jointly with the American Railway Engineers Association at the annual meeting of the American Water Works Association in Atlantic City, New Jersey, which dealt with water treating problems of particular importance in locomotive operation. In June the committee co-sponsored a program at the annual meeting of the American Society for Testing Materials in Detroit on corrosion problems in boiler operation. Also in June the committee co-sponsored a program on turbine blade deposits and steam contamination at the semi-annual meeting of the American Society of Mechanical Engineers in Milwaukee.

At the 1948 annual meeting of the American Society of Mechanical Engineers in New York, the Joint Research Committee cooperated with the Power Division on a program concerned with corrosion problems in high pressure boilers and devoted particular attention to discussion of hydrogen embrittlement.

A general reorganization was completed toward the end of the year in the subcommittees of the Joint Research Committee. Much of the work, for which the earlier subcommittees had been organized, has been completed and the need for their particular functions has therefore disappeared. The new organization provides for subcommittees as follows:

*Subcommittee on Deposits*
Chairman—Mr. R. C. Bardwell,
Superintendent of Water Supply,
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Hall Laboratories, Inc.
Pittsburgh, Pennsylvania.

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Chairman—Mr. W. L. Webb,
American Gas & Electric Service Co.,
30 Church Street,
New York 8, New York.

The subcommittee chairmen are now organizing the memberships of their committees and studying the problems which are of prime importance in the field under their jurisdiction. As a result of this work, it is expected that during 1949 the Joint Research Committee will sponsor further society activities and papers on various phases of boiler water treatment and may suggest some research problems to which the committee should devote particular attention.

# GAS TURBINE POWER PLANTS—A review of the study and application of the gas turbine to marine propulsion.

BY SEYMOUR COMASSAR.

THE AUTHOR.

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He served as Assistant Engineer Officer of the U.S.S. *Farragut* (DD 348) from January 1944 to October 1945 and as Engineer Officer of the U.S.S. *John R. Craig* (DD 885) from November 1945 to June 1946.

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*Introduction.*

The first patent covering a gas turbine was granted to John Barber in 1791. The sketch of this patent included all the elements of the modern gas turbine except that the compressor was of the reciprocating type. In spite of this early start, up till little more than ten years ago, the gas turbine appeared to be no more than a Utopian scheme for the future. Now however, engineers feel convinced that gas turbine power plants can offer serious competition to existing power generating stations of various kinds. The recent rapid development of the gas turbine is due to progress in the metallurgical field, which greatly increased its practicability and makes available a new prime mover capable of higher economies, in the power field.

The possibilities of the gas turbine in the field of marine propulsion have aroused the interest of marine engineers, and their participation in this development is concrete proof of the break in the traditional conservation in marine engineering design. From the work being done in the application of the gas turbine to marine propulsion, it is evident that marine engineers will no longer follow by a span of years the practices ashore but will give direction to some of the future innovations on land. Specific marine propulsion units utilizing the gas turbine, which illustrate this initiative in development, will be described later in this paper.

*Marine Power Plant Requirements.*

Marine power plant construction and operation requirements differ from stationary plants in many respects. Primarily, marine plants have necessity for rapid maneuvering over a wide range of loads. This necessitates the ability of the plant to respond rapidly to changes in speed. This situation arises when the vessel is maneuvering in harbors and when underway in the open sea. The requirement is even more pronounced in the case of naval vessels than it is for merchant vessels.

Provision must also be made in marine plants for astern operation, which requires reversal of rotation of the propeller but may or may not demand reversal of the prime mover, depending upon the transmission arrangement utilized. This factor will be treated in more detail later in this discussion. Astern operation is required in docking and undocking and in many maneuvering as well as emergency operations of both merchant and naval vessels.

Size and weight of marine power plants must be given more careful consideration than that of stationary plants, since the economics of cargo and armament space utilization is of prime importance in the necessarily limited volume of a vessel. The economy of operation is also directly related with this consideration, since the space to be alloted for fuel storage too is limited. The range of operation of any vessel is a factor that is constantly trying to be improved upon, since the longer a ship is able to steam between refuelings, the more self sufficient it can be and the less vulnerable is it to be a non-paying investment. The requirement of long range of operations between refuelings was forcefully shown in the last war for both merchant and naval vessels.

The necessity of carrying overload is obvious for naval vessels, which use full power less than 10% of their time underway. The cruising or standard speed of naval vessels is approximately 15 knots, while they must be able to develop a flank speed of approximately 25 knots in maneuvering operation. The demand for their full power runs, which is also called upon in emergency operation, is as high as 40 knots. This overload factor is not as important for merchant vessels, since 90% of their time underway is at nearly full load operation. This requirement makes it necessary for marine propulsion plants to have high part load efficiency characteristics.

The unavailability of repair facilities while underway and the general level of efficiency of the crews makes simplicity of the marine plant a greater requirement than in the case of stationary plants.

In summary, the factors mentioned above, which were rapid maneuvering, astern operation, size and weight, economy, overload and simplicity, make the design requirements of a marine propulsion gas turbine power plant much different than for stationary power plants.

*Gas Turbine Power Plant Cycles.*

A brief review of fundamental gas turbine power plant cycles will be given at this point for standardization of nomenclature and reference for processes referred to later.

The simplest form of gas turbine plant is one that operates on the open cycle and which consists of a compressor, combustion chamber (or combustor) and a turbine, as shown diagrammatically in Fig. 1. The compressor takes in atmospheric air and raises its pressure. In the combustor, fuel burns in this air, raising its temperature and increasing its heat energy. The fuel burned may be liquid or gaseous. This produces a working

fluid that is expanded in the turbine to develop mechanical energy. Part of the gas turbine's energy output goes to drive the compressor and the remainder is available as useful work. The expanded gases from the turbine, exhaust to the atmosphere.

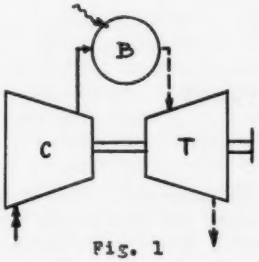



Fig. 1

A plant such as this operates with a thermal efficiency of approximately 20%, when the temperature of the turbine inlet gases is limited to 1300F. This temperature is limited for metallurgical reasons and the present stage of development of the gas turbine art has raised this limiting, constant temperature to approximately 1500F., with the use of various means of cooling the first stage of the turbine.

The efficiency of this basic plant can be increased in two ways without added apparatus or increased plant complexity. The first method is to increase the machine efficiencies, but due to the high levels of compressor and turbine efficiencies already reached, future prospects are necessarily limited. In both cases, the maximum useful efficiencies that have been utilized in experimental plants is in the order of magnitude of 85-88%. The second method of increasing the thermal efficiency is to raise the allowable gas temperature at the turbine throttle, as stated above. Constant metallurgical research and progress on this score makes the path promising. Calculations showing the increase in thermal efficiencies by use of these two methods are included in this paper.

The efficiency of the open-cycle plant can be increased without exceeding limiting conditions of machine performance and materials, but at the cost of increased plant complexity, by adding a regenerator as in Fig. 2, adding either

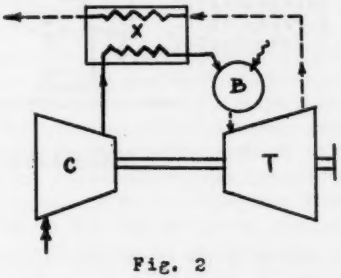



Fig. 2

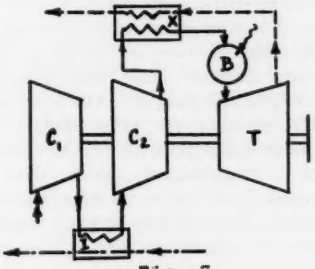



Fig. 3

intercooling or reheating processes in addition to the regenerator as in Fig. 3 and Fig. 4, or by adding both processes

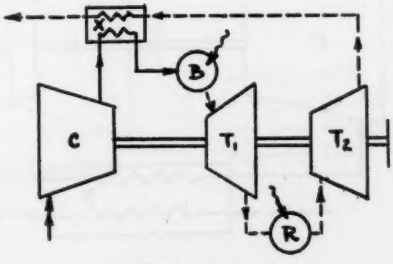



Fig. 4

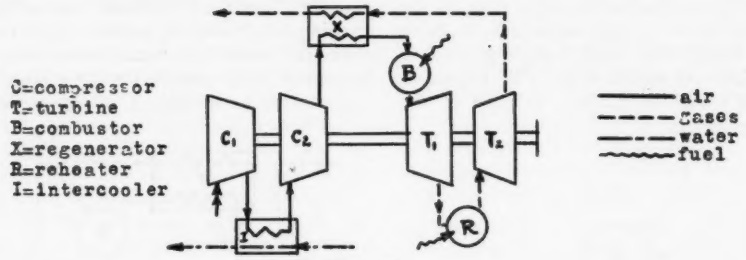



Fig. 5

OPEN-CYCLE GAS TURBINE PLANTS

in addition to the regenerator as shown in Fig. 5. Addition of intercooling or reheating without use of a regenerator lowers the cycle efficiency, but improves efficiency of the compressor and the turbine. Addition of a regenerator permits reclaiming of some of the heat of the turbine exhaust gases that would ordinarily be discharged to the atmosphere.

The closed-cycle plant differs from the open-cycle plant in that the medium used is recirculated, rather than discharged from the cycle after use as in the case of the open-cycle plant described above. As shown in Fig. 6, air passes through the compressor, where the pressure is raised, and then passes through an air-heater, where its temperature is raised by indirect heating. This method of transferring heat to the air allows it to pass through the cycle without being contaminated by the products of combustion. The air expands through the turbine in the usual fashion and then passes to a precooler, where the air is restored to its original temperature before it again enters the compressor. The use and results of regeneration, intercooling and reheating are similar to the open-cycle.

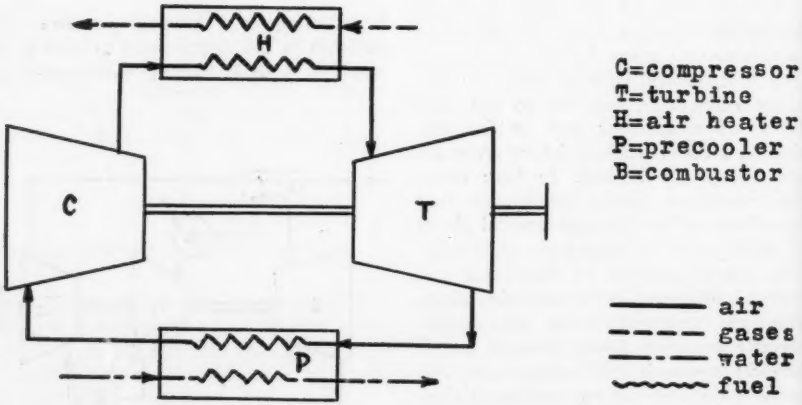



Fig. 6 CLOSED-CYCLE PLANT

Most of the advantages of the closed-cycle over the open-cycle grow out of the fact that it operates at pressures above atmospheric throughout the cycle. The compressor inlet pressure could very well be five atmospheres, which would produce correspondingly greater pressures throughout the system. These higher pressures mean higher air, or other medium, density and smaller volume, hence smaller dimensions for the machines. In turn, smaller dimensions permit higher temperatures without exceeding stress limits. High density also improves heat transfer coefficients and this, coupled with smaller volumes, makes for substantial reductions in size and weight of heat exchangers and may perhaps balance out the fact that equipment had to be added to make a closed cycle plant. Fuel flexibility is obvious since the air is heated indirectly and the products of combustion do not become a part of the medium flowing through the system. Plant output may be changed by raising or lowering the working medium density, which does not change temperatures and hence gives good part-load efficiency.

A semi-closed cycle for gas turbines is shown in Fig. 7. This cycle burns fuel directly in the circuit in a conventional open-cycle type combustor but is primarily a closed-cycle in that the greatest part of the working medium is recirculated. A continuous supply of fresh air is drawn into the system by means of the small compressor, which air serves to support the combustion process, while the effect of increasing the amount of working fluid in this manner is taken care of by operating the small compressor drive-turbine on the working medium of the system, and exhausting this turbine's expanded gasses to the atmosphere. Direct firing, in this type of system, eliminates the large air heater required for heat transfer in the closed-cycle system but fuel flexibility is lost. The high pressure operation however, still results in appreciable size savings.

Other variations of closed and open-cycle plants have been suggested and built and one other type of semi-closed cycle will be described in the section of this paper entitled Description of Plants.

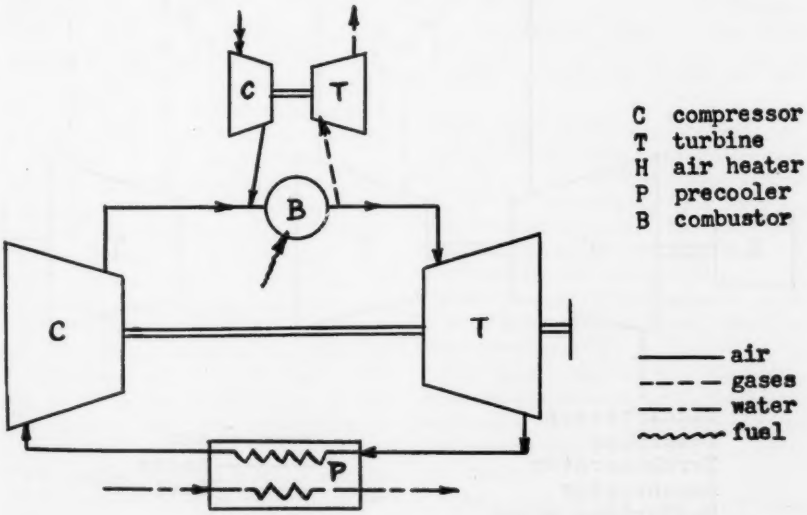



Fig. 7 SEMI-CLOSED CYCLE PLANT

The fundamentals of the gas turbine art have been described and illustrated in this section and will serve as reference in the later discussion of actual marine gas turbine plants, which have been derived on the basis of these fundamental plant approaches.

*Description of Plants.*

The interest of marine engineers in the development of the gas turbine is shown by the comparatively large number of designs and units that have been actually built for work in this field. Several of the larger and more important units will be described and pictured in this section.

One of the largest plants built for application to marine propulsion is the 3500HP. unit, which was constructed by the Allis-Chalmers Manufacturing Co. for the U. S. Navy. This plant was installed in 1944 at the Naval Engineering Experiment Station at Annapolis, Md. and has been undergoing a series of tests ever since to collect data on gas turbine plants for future use in design and construction of marine, stationary and other land use plants. A diagrammatic layout of this plant is given in Fig. 8.

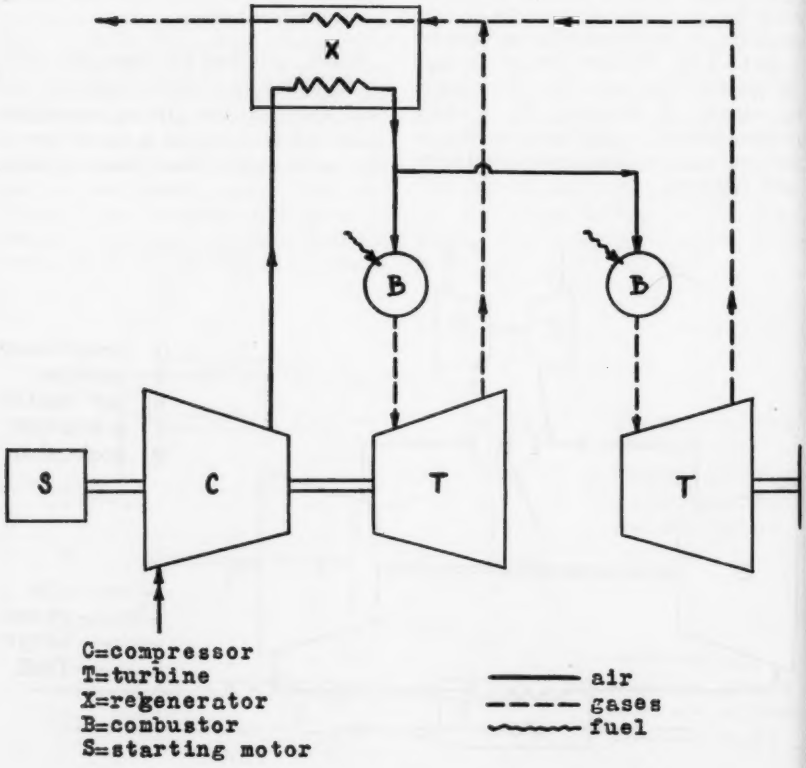



Fig. 8 ALLIS-CHALMERS MARINE GAS TURBINE POWER PLANT

The cycle selected for this experimental plant is an open-cycle similar to that shown in Fig. 2 but more complex in that a two-shaft arrangement is utilized, which is a result of the plant's eventual application to marine propulsion. This plant utilizes a parallel turbine arrangement, one turbine driving a compressor and the second or output turbine driving a dynamometer. In an actual installation for propulsion, the output turbine would of course drive the propeller.

Air enters the axial-flow compressor at atmospheric pressure and is compressed to a pressure of four times atmospheric. From this point the air flows through a regenerator where it is heated to approximately 750F. and from this apparatus, the working fluid flows to two combustion chambers, one of which supplies the compressor turbine and the other supplies the output turbine. The air flow to the power turbine combustion chamber may be regulated with a gate valve installed in this line so that it is possible to independently control the power generated by each of the turbines. The gases, after expanding through the turbines, then pass through the regenerator to heat the combustion air. From the regenerator, the gases are exhausted to the atmosphere.

The design and construction of the components of this plant are of worthwhile interest and a brief discussion of these will be given. The compressor used is a twenty stage airfoil-section machine, which runs at very high efficiency, of the order of 85%, when compressing the full load volume of 40,000 cfm of free air with a compression ratio of four. The simple character of this compressor and the absence of gears, sliding rods, or pistons are favorable factors in the direction of reliability, quiet operation and lack of vibration. It will be seen later, in the discussion of the Elliot plant, that this factor is not as favorable in that unit as it is for this Allis-Chalmers plant.

The regenerator used in the Annapolis unit is of the counterflow heat exchanger type with a heating surface of 8500 sq. ft. and has an effectiveness of approximately 60%.

The two combustion chambers utilize mechanical atomization to promote rapid burning of the No. 2 furnace oil that is used. It is predicted that fuels as low in grade as Bunker C will be able to be used in similar installations. The form of the combustor and the use of an air cooled flame tube make the use of brickwork unnecessary, hence the unit will respond quickly to load changes. In a service installation, the fuel control could be reduced to two manually or automatically operated control valves. It is seen then, that this part of the plant is comparatively simple and has the ability to respond to rapid maneuvering, which are prime requisites of marine propulsion power plants.

The two turbines of this plant are essentially the same in construction. Each has five pressure stages and operates at 5200 rpm at full load. The first two stages of the turbines utilize impulse blading and the last three stages have conventional reaction type blading.

The unit can be started with less than 100HP., and an electric motor is connected to the compressor unit by a clutch for use up to a starting speed of approximately 25 to 35% of full load speed. After the fuel is ignited in the combustors, the starting motor is shut down (disengaged) and the unit is brought up to speed of operation by manual fuel control.

Being an entirely experimental unit, this Allis-Chalmers plant was constructed and laid out for flexibility of operation and ease of taking data rather than for high thermal efficiency. The tests that have been run on this unit have been for progressively increasing turbine inlet temperature and the last reported series of runs were made with an operating temperature of 1450F.

This plant, while it is a land unit, shows the practicability of marine power plant units, since it was designed for that ultimate application, and is a practical reality.

Another gas turbine plant for marine propulsion was built for the U. S. Navy by the Elliott Co. and is set up and operated by this company's own facili-

ties. The general arrangement of this plant is shown in Fig. 9.

This plant too is a two-shaft arrangement but uses two compressors and utilizes intercooling and reheating. Basically, this plant is similar to Fig. 5, which shows a one-shaft arrangement utilizing regeneration, intercooling and reheating. The Elliott plant is also an open-cycle arrangement.

where it is heated by the exhaust gases, and then passes on to the high-pressure combustion chamber, where it is heated to 1230F. before entering the high-pressure turbine. The air expands to a pressure of approximately three and a half times atmospheric in passing through this turbine and then enters the low-pressure combustion chamber or reheater where its temperature is raised to 1207F.

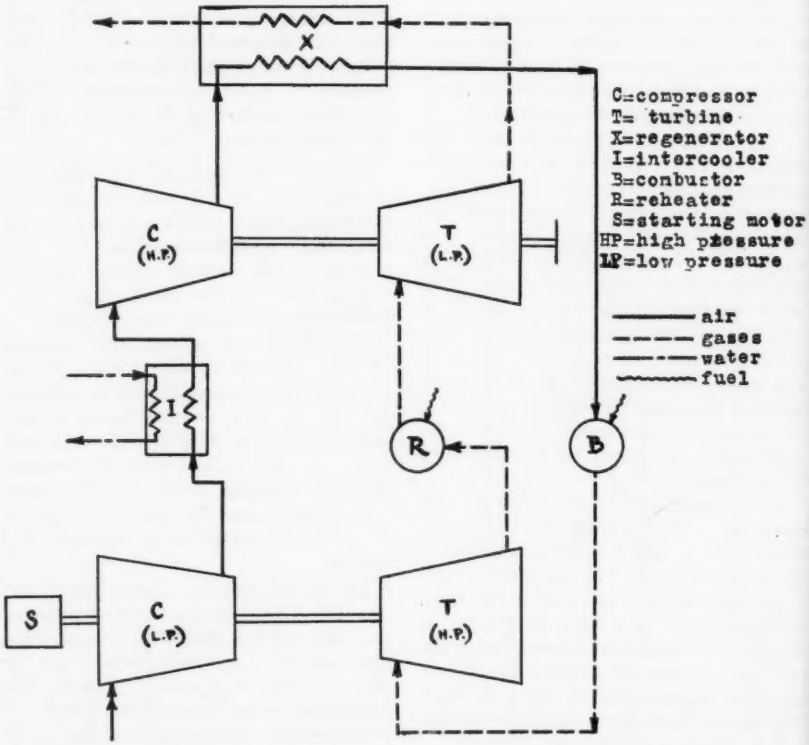



FIG. 9 ELLIOTT CO. MARINE GAS TURBINE POWER PLANT

The flow of air in this unit begins at the low-pressure compressor, which takes in atmospheric air and compresses it to a pressure of three times atmospheric. The air flows through the intercooler, which lowers the temperature, and then through the high-pressure compressor, which compresses the air to six times atmospheric pressure. From this point the air flows through a regenerator,

From this combustor, the gases then expand through the low-pressure turbine and pass through the regenerator and exhaust to the atmosphere. 5000HP. is realized from the low-pressure turbine, half of which goes into driving the high-pressure compressor and the remainder of which would be utilized in driving the ship's propeller.

By reference to the Allis-Chalmers plant layout, it is noted that for a two-shaft arrangement it is not necessary to use two compressors, but for a marine installation it is a desirable feature, since individual compressor drives make for better part load efficiencies. This is a necessity for naval vessels, which as previously noted, operate at a small percentage of their full power most of the time. This feature is not as important for merchant vessels.

Best efficiencies are realized when the temperature of the compressed air is as high as possible before it enters the turbines. Thus for best economy, it is desirable to operate at reduced power by decreasing the supply of air rather than by reducing the gas temperature. The best approach to high temperature, reduced flow operation is realized in this type of plant because the components are so arranged that the main power, or output, turbine can always be operated at full temperature. Complete control is obtained by regulating the fuel flow to the turbine driving the low-pressure compressor. Since the amount of air that enters the system is controlled by this compressor, it is apparent that this one feature can produce ease of control and at the same time, permit efficient operation of the power turbine.

The Elliott plant utilizes Lysholm positive displacement compressors, which have capacity control independent of compression ratio. These rotary positive displacement compressors are slightly lower in efficiency than centrifugal compressors and have lower capacities but are always stable, adjusting their discharge by regulating the speed of rotation. This leads to higher part load plant efficiency.

The operating data of the Elliott plant is of interest and some of the more important points will be noted here. The two compressors have identical rotors throughout, the low-pressure unit being double and the high-pressure unit being a single unit. The capacity of the low-pressure unit is 25,000 cfm and both compressors have a built in compression ratio of 3. The feature of a "built-in" compression ratio is peculiar to Lysholm compressors, which by virtue of their construction, cannot change this ratio, appreciably.

This plant can be maneuvered rapidly being capable of a change from light load to full load in 15 seconds. From full speed ahead of 3000rpm. to a no-load speed of 1400rpm. and then back to the 3000rpm. speed requires but 90 seconds.

Theoretical studies of this cycle in 1943 showed that a thermal efficiency of 32-33% would be obtained, but in practice the efficiency obtained was 29% at full load and 20% at one-quarter load. This was due to the pitch of the turbine blades, which were designed to be satisfactory for steam as the working fluid but were too large when using the gases of the cycle. This matter can be remedied easily in further construction however, and so a thermal efficiency of 33% can be obtained in this type of plant.

A warming up period of 2-3 hours is required for the plant at present, when starting from the cold condition but in future construction this disadvantage can be eliminated by more careful consideration of thermal expansion.

A great deal of objectionable noise accompanies the use of the positive displacement compressors and at present there is no satisfactory solution to this problem.

In spite of the above mentioned factors, this plant is a successful one and is one more step in the development of a practical marine propulsion gas turbine power plant.

Still another marine plant has been built, this one by Sulzer Bros. A layout of this plant is shown in Fig. 10.

It will be noticed that the cycle of this plant is a combination of the open and closed-cycles, but is different from the semi-closed cycle shown in Fig. 7. The closed-cycle of this arrangement, which handles clean air, is composed of two compressors and their drive turbine, a regenerator and a precooler. Fresh air enters the system from an auxiliary compressor ahead of the precooler. The heater combines direct and indirect firing and burns the fuel in the air drawn from the cycle ahead of the heater. Products of combustion pass through the heater tubes giving up heat to clean air flowing over the tubes. Clean air supplies the compressor drive turbine and combustion products power the separate output turbine and exhaust to the atmosphere.

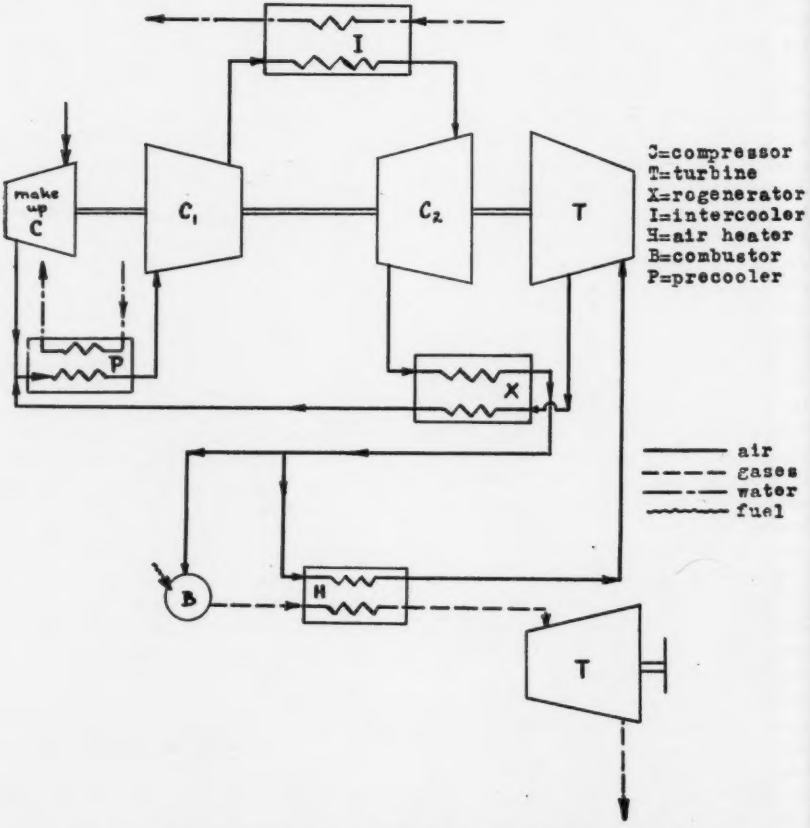



Fig. 10 SULZER BROS. MARINE GAS TURBINE POWER PLANT (SEMI-CLOSED CYCLE)

The turbines of this plant operate at an inlet temperature of 1200F. and the cycle pressures are four times atmospheric at the entrance to the first compressor and sixteen times atmospheric at

the discharge of the second compressor.

This plant arrangement gives all the size and weight advantages of the closed cycle and in utilizing a separate power turbine leads to high part load efficiency, which is required for marine plants.

Another open-cycle marine power plant using gas turbines is under design by DeLaval. This plant is a small experimental unit of 750HP. employing three shafts, as shown in Fig. 11. This cycle too, utilizes regeneration, intercooling and reheating. The three-shaft layout is provided to give a reversing feature to the plant, which is so necessary for marine propulsion.

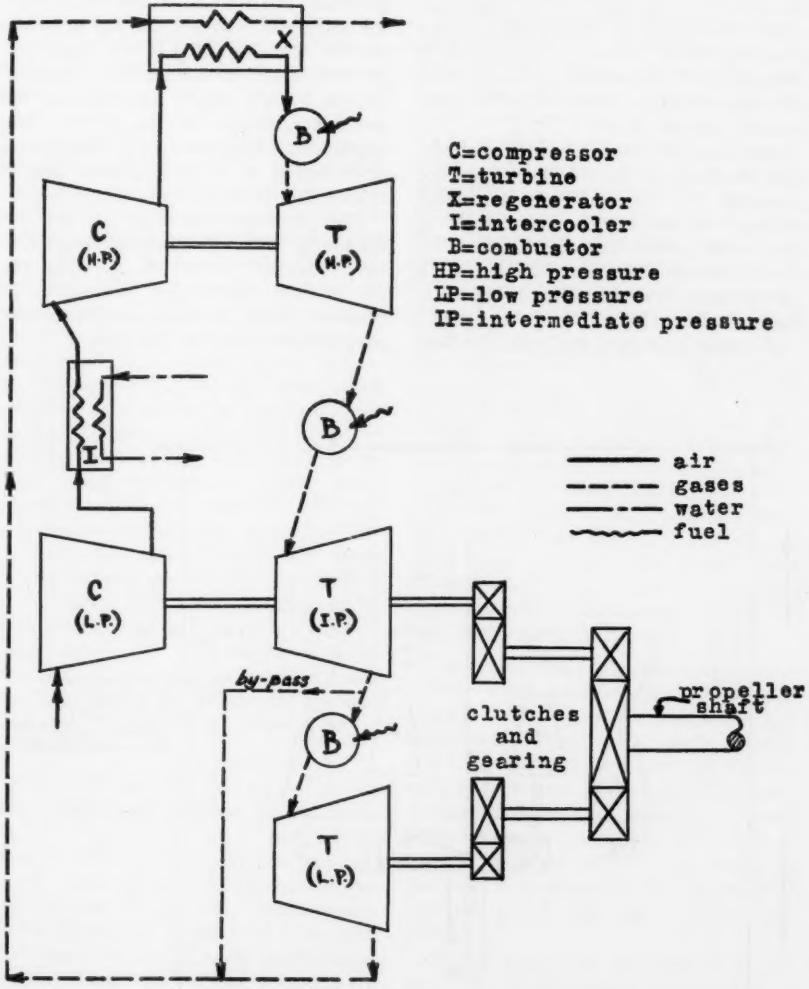



Fig. 11 DE LAVAL MARINE GAS TURBINE POWER PLANT

The method of operation of the plant is as follows: The low-pressure turbine, on a separate shaft, provides ahead power, being connected through a clutch and gearing to the propeller shaft. Under this condition, the intermediate-pressure turbine is declutched and drives only its own compressor. For astern operation, the low-pressure turbine is bypassed and declutched and the intermediate-pressure turbine, which operates with reverse rotation, is connected to the shaft by clutch and gearing. Additional energy resulting from greater pressure drop through the intermediate pressure unit provides power to drive both the compressor and the shaft.

One other marine propulsion gas turbine plant will be described to show the variation in concepts of propulsion methods in the development work to date. This plant is one designed by Brown-Boveri for installation in a British ship. The installation layout is shown in Fig. 12.

The plant is an open-cycle one developing 6000HP. and consists of two power turbines with ahead and astern sections. The turbines operate at 4300rpm. driving the propeller at 110rpm. through reduction gearing. The propeller is a fixed bladed type and the plant is very similar to steam installations in use at the present time. The gas inlet temperature to the turbines is 1112F. and the thermal efficiency of the plant is 26.5%. Consumption of Bunker C fuel oil is .53 lb. per BHP-hr. and the weight of the plant is 73 lb/BHP. A larger thermal efficiency can be obtained by use of larger regenerators but of course this increase is limited to the space available in the machinery spaces of the vessel. The engine room length with this installation is estimated to be 15% shorter than a similar Diesel propulsion installation.

The preceding examples of the development work that marine engineers have engaged in toward utilizing the gas turbine plant for propulsion is evidence of its ultimate acceptance and adaptability to marine propulsion.

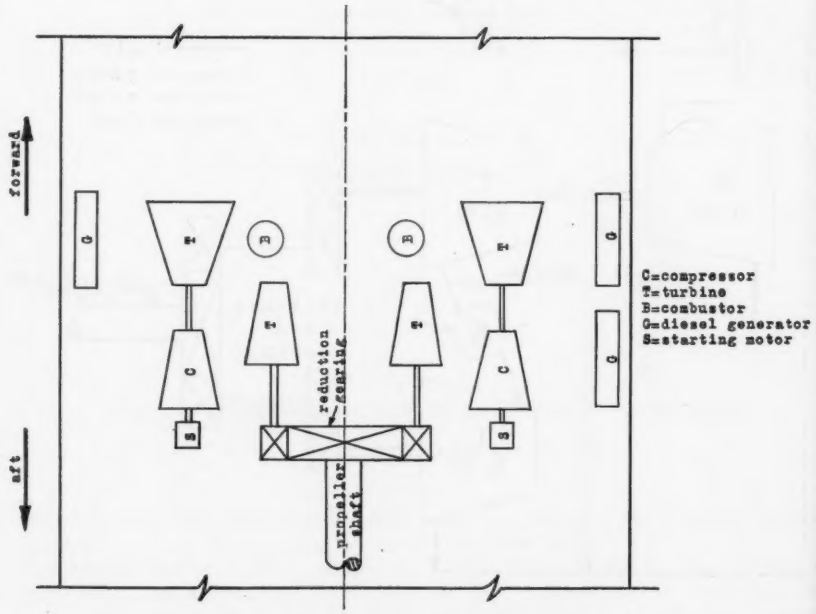



Fig. 12 BROWN BOVERI MARINE GAS TURBINE POWER PLANT

*Control and Part-Load Efficiency of Plants.*

The preceding section of this paper has given a discussion of marine gas turbine power plants generally and has also described specific plants. At this point it is necessary to study the control and part-load efficiency characteristics of the various plants in order to determine the extent to which they may be utilized successfully to meet the requirements previously mentioned for marine power plants.

As a basis for comparison of the more complex arrangements of plants that have been built and tested, the single turbine, simple open-cycle plant similar to Fig. 1 will first be considered. For this type of plant, the best partload efficiencies occur when the plant is operated at constant speed. When a regenerator is added to this cycle to reclaim some of the heat of the turbine exhaust gases, the best part-load efficiencies occur when the plant is operated at constant temperature. This type of control however, results in high turbine exhaust temperatures which necessitate costly heat-resistant materials. Another difficulty with constant temperature control is that at about 10% plant output, the compressor becomes unstable, having reached its "pumping limit". The single turbine plant can be used quite satisfactorily for direct drive ship propulsion at loads greater than 30%, and operation at loads less than 20% is not practically possible since the design—point temperature is exceeded.

For a dual turbine plant similar to the Allis-Chalmers plant or the Elliott plant, Fig. 8 and Fig. 9, there are TWO METHODS OF temperature control, namely equal turbine inlet temperatures and unequal turbine inlet temperatures. Either condition can be held by proper fuel feed to the combustors. In this manner the compressor turbine temperature may be maintained constant at all times and the power turbine temperature can be varied to suit load conditions. For either method of control however, there is found to be little variation in the part-load efficiency characteristics.

There are many considerations governing the selection of a particular method of control of a plant and some of the more important ones will be considered.

The life of a plant is affected primarily by "creep" of the material at elevated temperatures and by shock due to sudden changes in temperature. In order then to promote longer life, it would be desirable to operate at reduced temperatures at part-load if an acceptable efficiency curve could be obtained. However, when changing load under these conditions, distortion enters the picture and may cause mechanical failure.

The important concept of simplicity of plant is to be considered in deciding upon a method of obtaining high part-load efficiencies. If, as in the case of merchant ships, operation is always nearly at full-load, high part load efficiencies are not required. On the other hand, naval vessels are operated most of the time at a cruising load of about 10% of full load and here, part-load efficiencies that are high are desirable. The closed-cycle plant has the highest part load efficiency but is the most complicated. The dual turbine plant is not the best for high part-load efficiencies.

For rapid load changes, the single turbine plant seems to possess the advantage because of its simplicity but mechanical difficulties are likely if there is a large difference between the turbine inlet temperature at part and full-loads. A plant that operates at fairly constant temperature would satisfy this requirement. The single turbine plant would be the fastest to maneuver if operated at constant speed and had a satisfactory drive.

In the matter of starting up of the plant, the single turbine arrangement is naturally the simplest. The dual turbine plants require that the compressor side of the cycle be started first while the power side is either shut off or bypassed. Cur-

rent domestic practice advocates the use of a starting motor on the shaft but European practice, in many instances, makes use of compressed air in bottles to start the plant. A turning motor is necessary to keep thermal deformation to a minimum but if compressed air were used, this motor could be made a great deal smaller than if it had to develop a high starting torque on the shaft of the apparatus. Most merchant ships and all naval ships require a supply of compressed air for shop work and for torpedoes, pneumatic rams, etc. and a demand for compressed air for starting of the plant would not mean the need for additional equipment.

In the actual installation of a plant in a ship, the total power requirement of the vessel is a great deal larger than one gas turbine plant is capable of handling. In this case a multiple unit plant could be installed in which cruising powers could be obtained by using one complete gas turbine unit, while full-speed power could be obtained by using two or three of such units. This method would improve the part-load efficiency of the plant.

It is seen from the above, that every particular marine installation requires separate consideration to weigh the various design and operational factors involved.

*Transmission Systems.*

From the preceding material it is seen that satisfactory control for ahead operation of gas turbine propulsion plants is a practical possibility. A serious handicap of the gas turbines application for ship drive is its lack of adaptability for astern operation. In present steam turbine drives, the astern turbine is mounted on the same shaft as the high pressure turbine and during ahead operation, the astern turbine rotates in a vacuum. This arrangement for a gas turbine would mean that the astern turbine would have to rotate in a relatively dense atmosphere, which would result in high windage losses. Therefore, some other means of obtaining satisfactory astern operation is necessary to consolidate the thermal efficiency gains which could be obtained by the use of gas turbines for marine propulsion.

Since the characteristics of turbine type machinery provide optimum efficiency at high rotative speeds while the characteristics of marine propellers require low rotative speeds for optimum efficiency, it is necessary to utilize a transmission system that permits both objectives to be realized, for a successful marine application. In considering the choice of a satisfactory transmission system, it is necessary that the requirements of a marine transmission system be listed. The requirements are these: (a) It must be capable of reducing the rpm from those required by the prime mover to those required by the propeller, (b) It must be reliable and simple in operation within the capabilities of the crews that are available, (c) The transmission losses should be a minimum, (d) The first cost and maintenance cost should be commensurate with those of the prime mover, (e) The size and weight of the transmission system should also be commensurate with those of the prime mover, (f) If the prime mover is not reversable, then the transmission must provide this feature.

The suitability of various types of transmission systems for a gas turbine application will be given considering the systems now in use or those that have been suggested.

The direct-current electric drive has little to offer for gas turbine plant application because its first cost is high and its efficiency is low. The weight and space required are above those of other systems and its advantage of adjustable propeller speed independent of prime mover speed can be obtained in another

drive. Generally then, this drive is unsuited to gas turbine plants.

The alternating-drive system appears well suited to plants using a separate power turbine such as the Allis-Chalmers and the Elliott plants. Its cost and size are well within reason and though its maneuvering time and ability are not of the best, it is favorably regarded for marine applications.

The system using mechanical gears combined with a reversable pitch propeller seems most suitable for general use. Its cost, weight and efficiency are superior to most of the drives available and its maneuvering and reversing characteristics are superior to all of the other drives. The control of this transmission is simple since all operations are performed by one lever. The record of its experience in actual installations is very satisfactory on the basis of reliability and ease of handling.

The system utilizing mechanical reversing gears similar to that shown in Fig. 11 of the DeLavel plant offers attractive possibilities to plants using a separate power turbine. This drive's maneuvering and reversing characteristics are good and the weight, size and cost are suitable. The only doubt existant with this drive's application is whether or not it can be designed and built to handle larger plants than it has been applied to up till this time.

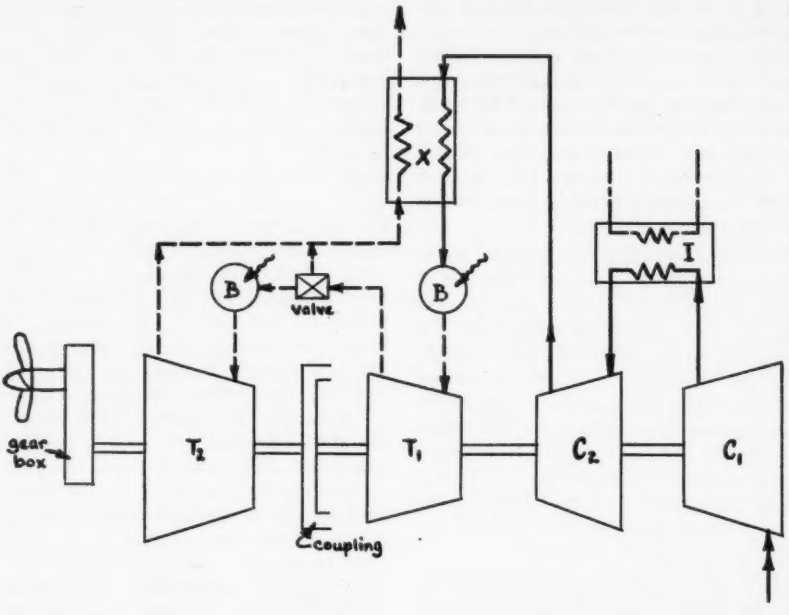
The possibility of achieving a satisfactory reversing gas turbine power plant appears improbable at this time. Because of the dense atmosphere that an astern turbine would rotate in during ahead operation, much additional equipment would be necessary for control of this turbine's atmosphere if this project were attempted. In view of the satisfactory transmissions available as given above, further development does not appear to be warranted on this score. One system of reversable gas turbine power plant is shown in Fig. 13. This method is outlined by R. Tom Sawyer in his book "The Modern Gas Turbine". The system consists of a power and a compressor turbine in line and connected by an electric coupling which is not energized when going ahead. The direction of rotation of the two turbines is opposite to each other. To reverse the vessel, the power turbine is bypassed and the electric coupling is energized, thus permitting the compressor turbine to drive the propeller in the reverse direction, through the power turbine. Study of this arrangement however, shows that the excess of power developed by the compressor turbine due to bypassing the power turbine is not sufficient to provide adequate astern power, which must be appreciable. This necessitates designing the compressor turbine with excess capacity over the compressor load. In normal operation then, the compressor turbine, which is the largest power component of the plant, will be operating at part-load. In addition, it would be necessary in reversing, to suddenly raise the operating temperature of this turbine so that thermal shock will become a consideration.

From the above considerations, it is once again evident that the peculiarities of the particular installation determine the selection of the proper transmission system for marine gas turbine drives. Generally however, the one drive that seems most destined to be used is the system using mechanical gears combined with a reversable pitch propeller.

*Possibilities for Future Development.*

In addition to the normal, continuing development in improvement of the various cycles and plants thus far designed and fabricated for marine gas turbine power plants, it is this author's opinion that several noteworthy paths of still further development are open in adapting the gas turbine to marine propulsion.

One path requiring further investigation is that in which the marine plant utilizes Lysholm rotary, positive displacement compressors. The advantage of use of such compressors over the other types available is the principal one




(Direction of rotation of $T_2$ is opposite to rotation of $T_1$)

C=compressor
T=turbine
B=combustor
I=intercooler
X=regenerator

air
gases
water
fuel

Fig. 13 REVERSABLE GAS TURBINE POWER PLANT

that this type compressor offers stable, high efficiency operation over almost any range of loads. The Lysholm compressor is capable of having discharge control independent of compression ratio. This feature, which results in high part-load efficiency is most desirable for marine plants, as was brought out in the previous discussion. At present, only one plant, the Elloitt Co. plant, makes use of this type compressor. Their application of it has been very successful and the only disadvantage found in its use, was the fact it was excessively noisy. This factor and the further application of such a compressor to gas turbine plants with the requirements peculiar to marine plants demands further exploration.

Another path of investigation open to marine engineers is in the application of the closed-cycle to marine propulsion plants. The advantages of higher full and part-load efficiencies of this cycle over the open-cycle, in addition to reduction in size and weight of the components of the plant offer attractive possibilities. It is true that the closed-cycle plant is more complex than the open-cycle one, but even this complexity appears to be less than that of a ship-board steam turbine plant installation. The requirement of cooling water supply for precoolers and intercoolers in the closed-cycle plant is of no importance for a marine installation, since a supply of cooling water is always plentiful, and

during ahead operation, little pumping power is required for circulation of this water. The waste heat available from the air heaters of a closed-cycle plant could be utilized in a waste heat boiler to generate steam for heating, operation of evaporators, cooking and the many other needs for steam or a source of heat required in a marine plant.

Still another development path open to marine engineers is the application of "wet compression" to a marine gas turbine plant. This process refers to the effect produced by compressing a mixture of finely divided liquid water suspended in the air supplied to the compressor, which is an effect similar to intercooling but is better in that no heat is removed from the cycle. The heat of compression of the air is absorbed as latent heat in the water, which is evaporated. At low compression ratios such as are now used in gas turbine cycles, the amount of water required is in the same order of magnitude as the weight of fuel used in the cycle, or 2 to 3% of the weight of air. This process reduces the work of compression and returns the heat removed from the air to the working fluid in the form of steam. This brings about, for a given size of compressor, an increase in the net power output from the cycle of as much as 20 to 30%.

Fig. 14 shows curves of efficiency of the Brayton cycle for wet and dry compression. In addition to the above, a further beneficial effect will be noticed from the curves of Fig. 14. Wet compression is seen to increase the cycle efficiency for a given compression ratio and also raises the compression ratio at which the maximum efficiency occurs and thereby again raises the thermodynamic efficiency. The practical result of this process is to give a marked reduction in fuel consumption and in first cost.

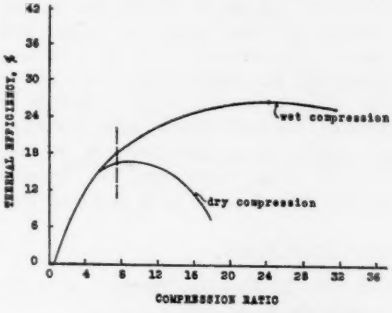



Fig. 14 EFFICIENCY OF BRAYTON CYCLE

Although no gas turbines designed for wet compression have been built as yet, tests have been run on a gas turbine designed for conventional compression operating with wet compression. The limited test results have borne out all of the above results based on calculations.

The problem of a supply of pure water in such a process should not phase marine designers since all marine plants require evaporators for production of drinking water and this small source of additional water requirement could be easily handled in the same apparatus. Present practice is to produce both drinking and boiler water by the same set of evaporators, merely decreasing the feed rate of the salt water to obtain the additional purity of feedwater over drinking water.

Future development of marine gas turbine plants in the broad development program of gas turbines in general then can lead the way to improvements in the entire field of design of such power plants.

*Comparison with Steam and Diesel Plants.*

In considering the three types of plants available as marine propulsion power plants, a brief comparison will be made on several desirable features necessary in such a plant.

On the basis of part-load characteristics, the efficiency of the gas turbine plant can compete successfully with both the Diesel and the steam plant. In fuel consumption, the gas turbine plant is in

a class with the Diesel plant and is very close in particular installations with the steam turbine plant. In addition, the gas turbine plant is able to use a cheaper fuel than Diesel oil and so has an advantage on that count. In weight and space requirements, the gas turbine plant has the advantage over both of the other types of plants. Athough at present, the gas turbine plant is limited in capacity to a point between the unlimited steam plant and the sharply limited Diesel plant, it is expected that very large gas turbine plants can eventually be constructed to overcome this partial handicap. Finally, it is expected that the gas turbine plant can and will approach the steam plant in reliability and low maintenance time and cost.

In conclusion, it is expected that once the obstacles associated with any new development of this kind are overcome, the gas turbine will be as suitable for continuous duty marine service as the steam turbine.

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# BOATS IN COLD WEATHER—Notes on factors to be considered and met in the operation of small boats under low temperature conditions.

BY COMMANDER ARTHUR C. BUSHEY, JR., U. S. N. R.

THE AUTHOR.

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The big problem incident to cold weather operation of boats is that of starting the engine. Other problems are: operation of boats in slush ice when fitted with sea chests; de-icing; protection of hull against thin ice; protection for propellers; and protection of personnel from the elements, or improvement of habitability.

Methods of cold weather starting of internal combustion engines have been numerous, but generally simple and of an improvised nature. The more prevalent practices are given below:

1. Providing additional electric power for cranking the engine. The practice of providing an additional storage battery connected in parallel with the boat battery works reasonably well if the booster battery is removed from the boat after lay-up and kept at a temperature of from 40° to 60° F. Batteries, dry and wet-cell, suffer a material reduction in output at sub-zero temperatures. The practice of providing low-cost, permanent installations with switchboard and ship-to-shore connections on boats to utilize external electric power for cranking the engine is likewise satisfactory where shipboard or shore-base low-voltage direct current of appropriate voltage is available.

2. Providing portable heaters for warming storage battery and engine crankcase. This method usually involves covering the machinery with a tarpaulin or temporary cover to prevent heat from being dissipated. It invariably presents a fire hazard. When electric heaters are used they pose again the question of external power, although the voltage is not restricted as it is in the previously described method. An installation of

recent date, which has been favorably reported, provides an enclosed electric heater of the rod type permanently installed in the crankcase.

3. Idling the engine when not in use. This method is satisfactory for short periods of continuous idling, but it becomes a chore for intermittent idling over long periods of non-use and eventually results in faulty operation of the engine due to carbon formation in the firing cycle.

4. Using heated oil in the crankcase before each boat operation. As this method requires draining the crankcase immediately after use and heating oil prior to use of the engine, the boat is never ready for emergency operation.

5. Using crankcase oils of much reduced viscosity or diluting the appropriate oil to make it light. Such oils are not suitable for sustained operation. Their chief use is for boats making short runs infrequently and in conjunction with the foregoing (4) to provide for emergency operation.

6. Using diethylene or a commercial starting fluid in the engine flame priming system. Permanently installed systems, simple in operation, have gradually replaced improvised methods. It is essential, however, to provide means of heating the electrolyte in the storage (starting) battery when the ambient temperature falls and stays below freezing. This method is at once highly efficacious and highly economical when used in conjunction with a combination of methods (1) and (2).

The Navy has adapted a commercial type of coolant heater for small boat operation in cold weather. The heater was designed for installation under the floor of buses to provide quick starting at a moment's notice after being parked outside on winter nights and at the same time insure a reasonable degree of habitability within the bus. It requires a space not greater than 31″ in length, 21″ in width and 14″ high. The unit is compact and consists of a control box containing switch and safety controls, a 2½ gallon tubular boiler, a power unit driving a set of breaker points, an enclosed fan and a fuel pump plus their associated controls to provide ignition, air and fuel for combustion; a stack for exhaust gases, a water inlet, an outlet and drains.

The heater is actuated by an aquastat installed on the engine side of the engine thermostat. A closed circulating water system, except for venting, is necessary for such a heater. With certain types of engine installations, it is necessary to change the wet type of exhaust to the dry type to fully accomplish a closed system. The heater unit is connected in series with the water system. The system is extended to include a pipe coil around the battery. In some installations it is desirable to include a coil around the fuel service tank, another around the crankcase, and to provide a radiator for habitability. The heater is entirely automatic. It has an OFF-ON switch; however the heater need not be turned off when the engine is in operation because the control devices are so accurate that they cut out the heater when the circulating water reaches a temperature of 145°F. The coolant heater consumes fuel, either gasoline or diesel, at the rate of ¼ of a gallon per hour and draws 8.5 amperes. Actually, fuel consumed is much less because service experience has shown that the heater is on intermittently. At first sight, this may appear expensive, but it is believed that an analysis of fuel saved by not idling the engine to maintain operating temperature and greatly reduced maintenance costs because of proper lubrication of the engine at all times will eventuate in the adoption of the coolant heater where there is a necessity for sustained cold weather operation of boats.

Operation of boats, which have a salt water cooling system, in slush ice requires special precautions to keep the sea chests and strainers clear. Slush ice filling the salt water intake results in loss of sea suction and overheating of engines. The simplest way to solve this problem is to eliminate it. This is accomplished

by having a closed fresh water cooling system with an external radiator or heat exchanger. The Navy is now using the keel type of heat exchanger. Recently, a patented rudder cooler appeared on the market. The use of an external heat exchanger also permits elimination of the strainer in the engine cooling system.

Winter grade lubricants and crankcase oil are necessary, regardless of the method of operation. Fuel should be as free as practicable of all traces of water or moisture to prevent ice clogging fuel lines, pumps and filters. To date it has not been found necessary to add an ice-preventative to the fuel to prevent formation of ice therein. Anti-freeze should always be added to the engine circulating water system during cold weather operations. The specific gravity of storage batteries should be checked daily and batteries kept fully charged. In sub-zero temperatures, engines should be turned over daily.

De-icing of boats is rarely attempted by a boat crew while the boat is underway. Exposed moving parts should be coated with a grease or an ice preventative which will permit easy removal of ice manually. Preclusion of the formation of ice on life boats when shipborne is accomplished by housing them in an enclosure which usually rests on the gunwale (a "Noah's Ark"), which can be heated electrically. The Noah's Ark must not interfere with lowering or hoisting.

Protection of the hull against thin ice is accomplished in the customary manner by metal sheathing of the underbody, usually with copper. Some replacement of sheathing is required annually. Propeller guards are provided to divert ice passed under the boat from the propellers. Although propeller casualties occur when pieces of ice jam between the guards and the propellers, frequency of repair is materially lessened when propeller guards are fitted.

In the selection of a boat for cold weather operation those with external rudder posts or outboard motors are not recommended as ice will form around the rudder post and interfere with steering. In the design of boats for this service it must be realized that at this time the only way to remove ice formed by freezing of spray and occasionally by green seas while underway is by hand labor on the part of boat personnel, a practice conducive to damage. It is considered impracticable in most boats to carry a heating boiler with sufficient capacity of either hot water or steam to thaw such ice. Electrical heating is likewise considered not feasible because of the practical limitations on output of electric power in boats.

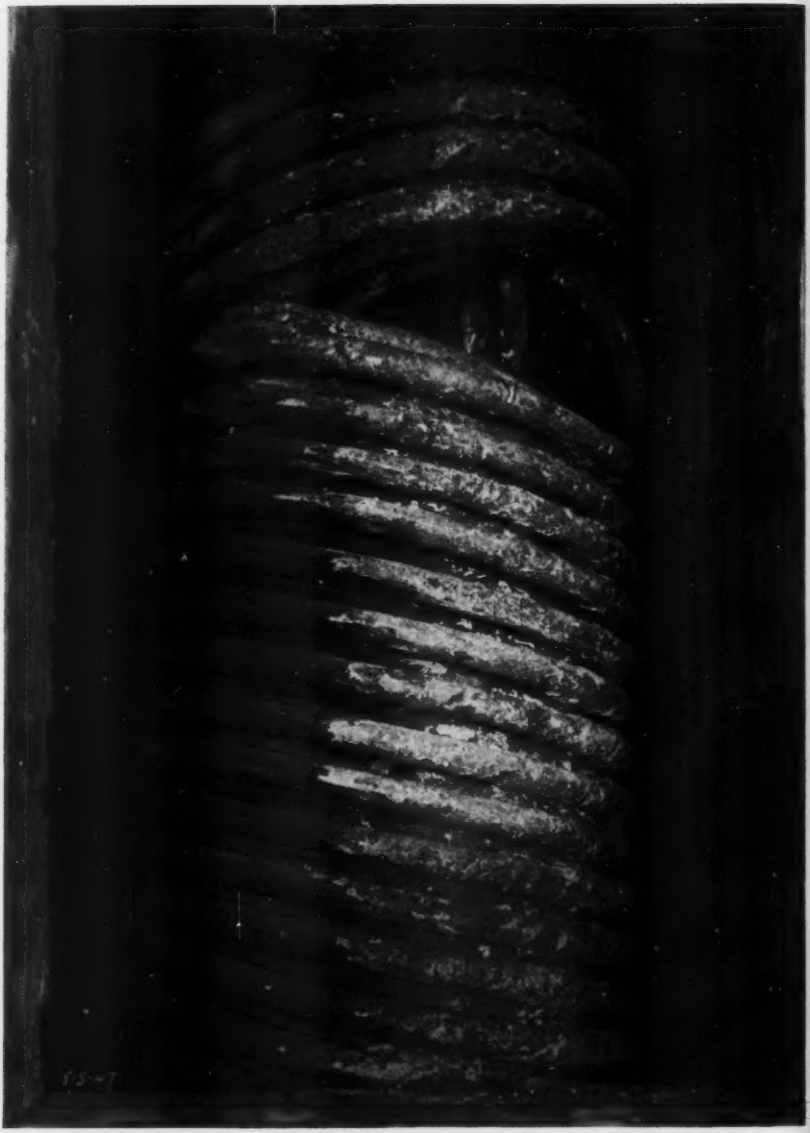

FIGURE 1.—Fireside deposit on the superheater tubes of a Naval boiler.

# FIRESIDE DEPOSITS—A study of minerals contained in fireside deposits of oil-fired boilers.

BY W. A. LAMBERTSON.

THE AUTHOR.

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*Introduction.*

Two articles have recently been published about causes and effects of deposits on the external surface of tubes in Naval boilers. In order to make more information available on the subject a study has been made of these deposits from the crystalline composition viewpoint. The results obtained are included in the report below. Figure 1, of a superheater tube bank, illustrates the nature of this deposit, which is called "fireside deposit" to distinguish it from the tube scale which often occurs inside the tubes. The combustion flow was from right to left whenever the superheater furnace, located at the right of the picture, was in operation. The fireside deposit is concentrated at the right where the hot gases first strike the tube bank, and it tapers off to the left. Deposits of a similar nature occur on the generating tubes, usually to a lesser degree.

*Review of Literature.*

In 1939 Letort (1) stated that Navy boiler fuel gases contain sodium sulfate and occasionally suspended iron. Reid, Corey, and Cross (2, 3) studied the fireside deposit in pulverized, coal-fired, slagtap furnaces. These authors identified ferrous sulfide, iron oxide, a solid solution of sodium and potassium sulfates, and small amounts of silica, alumina, and lime in the deposits. Hock (4) separated the fireside deposits into layers and stated that the major constituent of the tube-side layer (from analyses of two deposits) was vanadium oxide and the fire-side layer was sodium sulfate. McCloskey (5) indicated that fireside deposit was a mixture composed of from

> "one half to two thirds either of sodium sulphate or of a mixture of sodium sulphate and vanadium pentoxide; plus lesser amounts of the oxides of calcium, magnesium, silicon, iron, and aluminum; plus very small amounts of many other oxides."

In the most recent article on the subject Gray and Killner (6) identify sodium sulfate as the major compound of the deposits.

*Methods of Identification.*

Chemical analyses were made by the Naval Boiler and Turbine Laboratory in Philadelphia. The petrographic microscope was used in some identifications following the procedure as described by Rogers and Kerr (7) and others. Opaque phases were studied with the metallographic microscope as described by Farnham (8). The North American Philips X-ray apparatus was used with a Brown "Electronik" potentiometric recorder.

*Discussion of Results.*

Kracek and Ksanda (9) identified five polymorphous forms of sodium sulfate. At the temperature of the superheater tubes during steaming the stable form of sodium sulfate is the high temperature modification "I." When this is cooled to room temperature it may invert through some or all of the five polymorphous forms to modification "V" which is stable at room temperature. Some sodium sulfate "III" was identified with the petrographic microscope, but, because this form is unstable at room temperature, the determination was not made with 100 per cent certainty.

In all samples studied sodium sulfate constituted the most abundant crystalline phase. This is indicated by Table 1 which shows the normal range of chemical composition listed by the Naval Boiler and Turbine Laboratory. Sodium sulfate modification "V" was identified with the X-ray and the petrographic microscope. The X-ray results are shown in Table 2.

If sodium sulfate is exposed to the atmosphere for a sufficient length of time at temperatures below 90°F. it will combine with the moisture from the air and form a hydrate. The X-ray is not a posi-

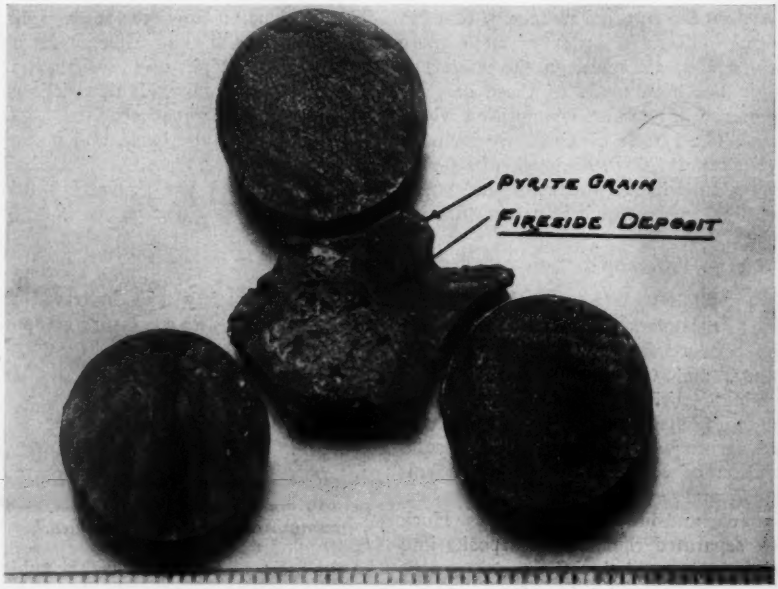


FIGURE 2.—A sample of fireside deposit from a Naval boiler. The relation of the deposit to the boiler tubes is indicated by the disks which represent cross-sections of tubes.

Table 1

NORMAL RANGE IN CHEMICAL ANALYSES OF FIRESIDE DEPOSITS.

| *Oxide* | *Percentage* |
|---|---|
| $SO_3$ | 40 to 50 |
| $Na_2O$ | 25 to 30 |
| $Fe_2O_3$ | 4 to 15 |
| $V_2O_5$ | 3 to 8 |
| NiO | 1 to 5 |
| $SiO_2$ | 1 to 2 |
| $Al_2O_3$ | 0.5 to 1 |
| MgO | 0 to 4 |
| CaO | 0 to 3.5 |
| $K_2O$ | 0.5 |
| $Cr_2O_3$ | 0 to 0.5 |
| $TiO_2$ | 0 to 0.25 |
| $P_2O_5$ | trace |

tive tool for identifying the hydrated form of sodium sulfate because in the process of normal sample preparation it tends to revert to the anhydrous form (9). However, hydrated sodium sulfate can be detected with the petrographic microscope.

The samples examined were kept in an air tight bottle and did not appear to be hydrated. Any hydration of the sodium sulfate in fireside deposit can only occur when the temperature of the deposit drops below 90°F. and therefore is not to be expected unless the boiler is shut down for an extended period of time.

The insoluble portion of the deposits examined was strongly attracted to a magnet. It was composed largely of opaque black crystals which were identified as magnetite with the metallographic microscope and the X-ray. Most of the iron in the fireside deposit appeared to be present in the form of magnetite.

A small amount of iron sulfide (probably less than one per cent) was identified as pyrite with the metallographic microscope. Figure 2 shows a sample of fireside deposit from a Naval boiler and indicates the manner in which it filled the space between three tubes. The pyrite shown on this sample appears as an almost invisible reflecting speck on the polished surface of a section of the deposit. Figure 3 is an enlarged picture of a pyrite grain. Because of the localized nature of the few pyrite grains found, the presence of pyrite is not to be

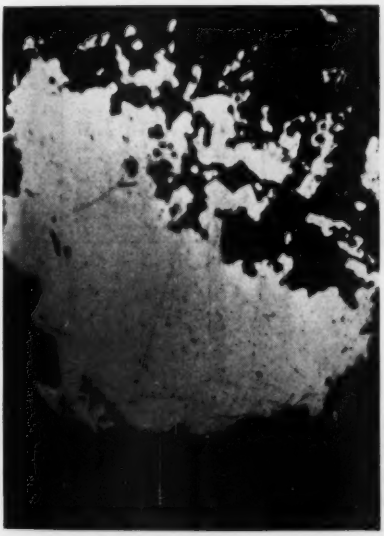

FIGURE 3.—Polished fireside deposit showing only the pyrite grain. 150X magnification with reflected light.

FIGURE 4.—Cross section of a sample of fireside deposit. 30X magnification with transmitted light.

taken as proof that the entire deposit was made under reducing conditions. It is suggested that more probably an occasional globule of oil will reach the boiler tubes, strike the deposit and burn, and reduce only the small amount of deposit with which it comes into contact.

Figure 4 is a magnified section of fireside deposit under transmitted light which shows white crystals of sodium sulfate on the "tube side" and some black magnetite on the "fire side."

TABLE 2.

X-RAY DATA ON FIRESIDE DEPOSIT

| Fireside Deposit | | Insoluble F. S. D. | | Magnetite (A. S. T. M.) | | Soluble F. S. D. | | $Na_2SO_4$-V (Kracek) | | $Na_2SO_4$-III (Kracek) | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| d | I | d | I | d | I | d | I | d | I | d | I |
| .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 4.76 | .2 |
| 4.69 | .4 | .... | .... | .... | .... | 4.73 | .5 | 4.67 | .6 | .... | .... |
| .... | .... | 4.24 | .2 | 4.21 | .1 | .... | .... | .... | .... | .... | .... |
| .... | .... | 3.89 | .4 | .... | .... | .... | .... | 3.82 | .2 | 3.94 | .7 |
| 3.71 | .3 | 3.74 | .4 | .... | .... | .... | .... | .... | .... | 3.77 | .6 |
| .... | .... | .... | .... | .... | .... | 3.64 | .3 | .... | .... | .... | .... |
| .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 3.48 | .4 |
| .... | .... | 3.29 | .2 | .... | .... | 3.27 | .4 | .... | .... | .... | .... |
| 3.19 | .2 | .... | .... | .... | .... | 3.20 | .5 | 3.18 | .6 | .... | .... |
| 3.12 | .4 | .... | .... | .... | .... | 3.10 | .4 | 3.08 | .7 | .... | .... |
| .... | .... | 3.03 | .3 | .... | .... | .... | .... | .... | .... | .... | .... |
| .... | .... | 2.98 | .3 | 2.98 | .6 | .... | .... | .... | .... | .... | .... |
| .... | .... | 2.91 | .3 | .... | .... | .... | .... | .... | .... | .... | .... |
| 2.79 | 1.0 | 2.80 | .4 | 2.79 | .4 | 2.80 | 1.0 | 2.79 | 1.0 | 2.80 | 1.0 |
| 2.66 | .3 | .... | .... | .... | .... | 2.66 | .4 | 2.65 | .6 | 2.63 | .8 |
| 2.52 | .4 | 2.53 | 1.0 | 2.54 | 1.0 | .... | .... | .... | .... | .... | .... |
| .... | .... | .... | .... | 2.43 | .3 | .... | .... | .... | .... | .... | .... |
| 2.34 | .2 | .... | .... | .... | .... | .... | .... | 2.33 | .3 | 2.37 | .7 |
| .... | .... | 2.09 | .7 | 2.09 | .7 | .... | .... | .... | .... | .... | .... |
| .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | 1.95 | .6 |
| 1.86 | .2 | .... | .... | .... | .... | 1.87 | .3 | 1.86 | .5 | 1.88 | .4 |
| 1.66 | .3 | .... | .... | .... | .... | .... | .... | 1.66 | .1 | .... | .... |

*Summary.*

The crystalline phase identified which was present in greatest amounts was sodium sulfate modification "V". Some sodium sulfate modification "III" was identified. Second to sodium sulfate was iron oxide in the form of magnetite. A small amount of pyrite was identified.

Although the major crystalline phases in this Naval boiler fireside deposit have thus been identified, a number of other crystalline and apparently glassy or amorphous phases were observed, but were not identified and are not mentioned above. In the samples studied these were present only in minor amounts.

*Acknowledgment.*

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NOTICE.

MEMBERSHIP LIST.

The Society has recently published and distributed a Membership List as of 1 February, 1949. In order that this might be available to members contemplating attendance at the annual banquet in time to be of assistance to them, publication was on the basis of information of record. It is intended as soon as practicable to furnish the membership cards on which each may show data desired listed and to prepare a new list, incorporating the information received. It is again desired, however, to emphasize the fact that report of changes in address, as occurring, is essential to prompt and accurate furnishing of the JOURNAL and other matter of interest.

# TESTING ELECTRICAL INSULATING VARNISHES—An exhaustive treatise in which all types of tests of electrical insulating varnishes are discussed as to their *method, variables* and *significance.* In two parts.

BY HERBERT G. STEFFENS.

THE AUTHOR.

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## PART I

**EDITOR'S NOTE:** The editor had three choices: to reject this treatise, to publish it entire to the exclusion of other subjects, and to serialize. The last was adopted. There is no logical breaking point. Mechanically the paper separates after the section "ELECTRICAL TESTS" and as a convenience that point was selected. The remainder of the paper embracing sections entitled: MECHANICAL TESTS, CHEMICAL TESTS and THE VARNISH SPECIFICATION will be published in the next, the August number of the JOURNAL. The entire BIBLIOGRAPHY is carried in this issue at the end of this Part I.

## INTRODUCTION.

Accurate estimation of production costs in the manufacture of electrical equipment presupposes sound knowledge of many factors in design, one of which is the performance characteristic of varnishes used as insulation.

The main purpose of an insulating varnish is to isolate conveniently and economically portions of the electrical circuit from one another and to exclude therefrom moisture, dirt, and other low-resistance materials. In addition to this, considerable emphasis is placed on the electrical, mechanical, and chemical stability of the varnish film under actual service conditions and during prolonged aging.

Attempts to satisfy these latter requirements have resulted in a considerable volume of work of highly technical nature with respect to the testing of insulating varnishes. In fact, it is because of this that varnish testing has been placed on a well-defined scientific basis.

The physical attributes of insulating varnishes, like many other materials, are defined by a series of operations and calculations. Therefore, exactness in varnish evaluation and comparison rests dependently on the philosophy and fundamentality of the test methods themselves.

Modern insulating varnish technology had its theoretical beginning in the

patents of Albert and Berend in 1911 relating to the modification of natural resins with phenolic compounds to form oil-soluble synthetics. It had its practical industrialization with the 100 per cent phenolics of Turkington and Butler in 1928. At some point in the interim there began a rapid development of test methods which were based on existent methods of that period, but which were specifically modified to describe these new materials and their associated properties.

The main purpose of this paper is to discuss the significance of the present methods in the complete definition of an insulating varnish.

## TYPES, USES, AND REQUIREMENTS OF INSULATING VARNISHES.

### *Types.*

The varied uses to which insulating varnishes are put preclude any possibility that one particular type will meet all requirements expected of it. Thus, insulation may be expected to bond high-speed rotating parts into one rigid and homogeneous mass, whereas in another part of the same apparatus inherent mechanical vibration may require that the insulation be flexible and resilient. Insulating varnishes may, therefore, be expected to group themselves into a variety of types, each predominating in some individual characteristics.

In general, insulating varnishes may be classified into baking and air-drying grades. Skinner (1) is of the opinion that baking grades were the first types of insulating varnishes used. This may be due to the generally superior flexibility and waterproofing characteristics of that grade. Air-drying types were apparently not more than moderately successful in fulfilling these requirements.

Baking varnishes are of two main classes: conventional or oxidizing and internal curing (deep-drying) or polymerizing. The former have been very popular in the past because of their ability to dry at relatively low baking temperatures to satisfactory degrees of resistance. The curing varnishes, on the other hand, are considered superior in that the higher temperatures required for complete polymerization also facilitate quicker production and repairs, and that the varnish sets to a homogeneous mass throughout and can be used to better advantage on high-speed rotating equipment, particularly at elevated operating temperatures. With exception of a few special characteristics, the performance qualities of each are comparable however.

Baking varnishes are usually of relatively high oil-resin ratio, termed "long length" in the trade. Metal driers added to catalyze the drying process are often of the liquid type or soft metal-soap type, added during the cooling part of the cook or after the varnish has been thinned. Rarely are driers of the oxide or hydrate type added during the cook except in cases where flexibility on extended heat aging is not required.

Air-drying varnishes are similar to the baking grades in composition except that there is contained a higher resin-oil ratio and higher drier concentration. Metal driers are usually of the oxide or hydrate types, such as litharge or manganese hydrate, and are added during the cook. Such compounding aids materially in shortening the drying period of the varnish. Solvents of low boiling range are used as thinners to insure more or less complete evaporation from the film at room temperatures.

Performance characteristics of air-drying varnishes are not of as high a quality as those of baking varnishes, since films of the former type require considerable time to develop their optimum properties.

Practice has been to manufacture a special type of air-drying varnish called "finishing varnish" which has exceptional film hardness and high gloss. Finishing varnishes are used to reduce dirt enlodgement and film breakage on

insulated parts. This type of varnish is, however, finding less use as the development of baking and air-drying varnishes progresses to include these characteristics in their general properties. The principal outlet of finishing varnishes at present is probably to satisfy a few more or less obsolete purchase specifications or for special applications.

Tests conducted at a Navy laboratory during the period 1943 to 1945 indicate that varnishes tested for compliance with Specification 52-V-13 fall into the following approximate frequency distribution for contract inspection tests:

Black baking (Grade BB)....30%
Clear baking (Grade CB).....24%
Black air-drying (Grade BA)..22%
Clear air-drying (Grade CA)..19%
Clear finishing (Grade CF)... 5%

Since the Navy is perhaps the largest single user of insulating varnishes of all types, these figures can be taken as fairly accurate relative demand ratings for the various types.

Each of the types of varnishes is generally supplied in either clear or black, although no entirely suitable reason has as yet been offered for the existence of these variations. The replacement of a part of the resin of the clear varnish type with a pitch or asphalt to make the black variety may, in some cases, contribute to a price differential. However, it is the author's experience that asphalts, while offered as relatively uniform in various physical properties, are quite variable in electrical properties, a characteristic which makes their inclusion in an insulating varnish somewhat objectionable from the standpoint of uniformity from batch to batch. Arguments are also used to the effect that clear types are specified in some instances where assembly code numbers marked on parts must be easily readable after impregnation and that black types are used where unsightly coil wrappings and winding errors are to be obliterated. Hazeltine (132) and Hepp (277) are of the opinion that black grades are possessive of superior dielectric strength, heat endurance, and moisture resistance, whereas the clear types exhibit better oil resistance. These assumptions are, however, at variance with the bulk of data reviewed by the author on thousands of tests conducted on varnishes of many different commercial sources. The author is inclined to agree with Hazeltine's later comments (147) regarding the use of clears and blacks as having become largely one of personal choice.

*Uses and Applications.*

Baking grades of varnish are used chiefly to impregnate motor and generator coils, relay assemblies, and transformer windings where a highly resistant and long-lived insulation is desired. Air-drying varnishes are used where shop facilities would preclude the use of baking grades, and where a relatively harder but not necessarily more flexible finish is required to reduce the danger of injury to the film due to mechanical shock and abrasion.

The insulation of a large winding with varnish is usually effected in about seven steps, as follows:

(a) singeing the coil (if cotton-covered) and pre-heating at about 100 degrees C. for a period depending upon the size of the piece, but usually not less than three hours,

(b) placing the piece while hot in an autoclave equipped with a vacuum pump and evacuating to about 28 inches of mercury until the piece is fairly cool,

(c) allowing varnish at a predetermined consistency to cover the piece and releasing the vacuum,

(d) applying air pressure of about 50 to 100 pounds per square inch to the surface of the varnish,

(e) removing the piece and draining at room temperature,

(f) baking the piece in the direction of drain at a temperature and for a period of time recommended by the manufacturer of the varnish,

(g) flash-dipping (rapid dip and removal while hot) the piece in reversed direction, withdrawing from the varnish and draining and baking as before.

Impregnation may also be effected by simple preheating and dipping without the use of vacuum. There is some strong disagreement to using vacuum to force the varnish deep into the piece with subsequent attempts at complete removal of all residual solvent within the piece.

Impregnating shops are generally equipped with dipping tanks set into the floor and extending up about one to two feet above floor level into which are placed the varnishes to be used in treating apparatus. These tanks are also equipped with removable covers to keep out shop dirt during periods when impregnation is not in progress. Draining of impregnated pieces is usually effected over the dip-tank unless the operation is one of chain-production using a conveyor mechanism. Larger shops are also equipped with facilities for filtering the varnish, making it possible to remove metal filings, dirt, gel particles, skins, lint, and other debris.

Baking is usually done in either gas or electric ovens of the gravity convection type or in infra-red chambers. In the latter case, some arrangement is usually necessary to insure equalized temperature about the entire piece, since the temperature developed in an absorbing medium is dependent on the square of the distance from the source of radiation, and also on the angle of the incident radiation.

Some large shops are equipped with forced-draft ovens which insure adequate circulation and ventilation, and promote more rapid drying of the varnish than the gravity convection type.

*Requirements.*

Emley and Tuckerman (200) state the principles governing the general requirements or values of an insulating varnish in the following order: "(a) the suitability of the article to perform the service expected of it in a satisfactory manner, (b) the durability of the article when used for that purpose, and (c) such requirements as to appearance as the user may demand and be willing to pay for."

Requirements which the author believes should be met by all insulating varnishes to some degree are as follows:

(a) suitability for application by dipping or spraying
(b) impregnation efficiency
(c) short drying or baking time
(d) moisture resistance
(e) high dielectric strength
(f) high insulation resistance
(g) low power factor
(h) oil resistance
(i) heat endurance
(j) elasticity
(k) abrasion resistance
(l) flexibility
(m) bonding strength
(n) impact resistance
(o) adhesion
(p) non-corrosive to metals
(q) resistance to ionization
(r) acid resistance
(s) alkali resistance
(t) non-oxidizing effect on transformer oils
(u) smooth and glossy film
(v) film stability
(w) dip-tank stability
(x) hiding power (opaques only)
(y) non-flammability
(z) non-injurious solvents

SUITABILITY OF APPLICATION. Varnishes should be suitable for application by spraying as well as dipping. Most oleoresinous varnishes, except perhaps certain pigmented types, are applicable by both of these methods. Although coating is almost always done by dipping, certain chambered assemblies cannot be satisfactorily coated in this manner, and spraying must be resorted to.

IMPREGNATION EFFICIENCY. The impregnating characteristic of a varnish is not an independent property. The high degree of polymerization during the varnish cook, which is necessary for good electrical and mechanical strength, may not permit thorough and complete impregnation or penetration of dense paper or cloth. For example, if the varnish is cooked almost to gelation, the resultant large molecule size may not allow adequate penetration into the paper

or cloth with which the piece may be primarily wrapped, or may not allow penetration between tightly wound layers of wire.

SHORT DRYING OR BAKING TIME. Rapid drying provides convenience in application and enables materials to be handled and relocated in apparatus shortly after impregnating. However, as pointed out by Harvey (101), rapid drying is usually indicative of low elasticity and durability, particularly on aging. Also, in cases where some specifications require testing of specimens after conditioning periods which are multiples of the measured drying time, short-drying varnishes do not attain the degree of resistance that slower drying varnishes do, and are therefore at a disadvantage.

MOISTURE RESISTANCE. Moisture is considered the one greatest factor in the deterioration of varnished insulation (1), (3), (30), (132), (244), (277). Weber (43), for instance, states that "a varnish film which turns white on immersion in water is of no value for electrical purposes". Stoppel (40) also concurs in this opinion, believing that absorption of water in relatively large quantities plays an important part in the destruction of the varnish film.

HIGH DIELECTRIC STRENGTH. Undoubtedly, the main purpose of insulating varnish is to increase the dielectric strength of the apparatus or primarily-insulated equipment to which it is applied. Many installations, such as newly-mounted motor and transformer windings, possess a fairly high dielectric strength but cannot maintain this under all conditions of service and aging, whereas the application of varnish to these elements raises the dielectric strength and maintains this level under severe service conditions. Some varnishes (243) are specially designated to maintain very high dielectric strength at creased edges, an unusual requirement.

HIGH INSULATION RESISTANCE. The American Society for Testing Materials (283) does not consider a material an insulator which possesses a resistivity of less than one megohm-centimeter. The Navy Department (295) is of the opinion that a varnish film of two mils thickness should show a resistance of at least one megohm after 240 hours immersion in distilled water, which, according to the Navy test, is equivalent to a resistivity of about 4000 megohm-centimeters. Insulation resistance is theoretically related to dielectric strength, although the usual errors in both these measurements would probably make experimental verification almost impossible, particularly in the presence of moisture. It is, however, considered by Mathes (296) that insulation resistance measurements serve as only an indication of other properties, and that the magnitude of the resistance is of no practical value except perhaps in high-resistance electronic circuit insulation.

LOW POWER FACTOR. A low power factor indicates a high degree of non-polarity of the dried varnish film, and suggests that the film possesses a suitable degree of chemical stability as well as low heating losses in alternating current fields. Except in specific cases where the varnish is to be used in power transmission where the problem of heat and power losses is a consideration, low power factor in itself is of little commercial significance in insulating varnishes of the impregnating type.

OIL RESISTANCE. Almost all insulating varnishes come into contact occasionally with oil, either deliberately as in oil-filled transformers, or by accident as in motor-splashings. Most oil, if clean, possesses good electrical insulation properties. However, films of varnish which are not adequately resistant to oil may lose toughness and continuity, particularly on continued exposure at high temperatures. Vogel and Narbutovskih (245) point out that hot-spot temperatures in some oil-filled winding assemblies may reach 140 degrees Centigrade within a short time after the apparatus has started cold. Oil resistance, is, therefore, considered a basic property of insulating varnishes.

HEAT ENDURANCE. Heat endurance is a measure of the ability of a varnish to

resist the natural embrittling and deteriorating effects of extended heat aging. The property of heat endurance is important from the standpoint of thermal expansion during large temperature variations under which the insulation must maintain its elasticity and continuity.

ELASTICITY. Retention of elasticity, or more accurately of the modulus of elasticity, during thermal and mechanical shock is a necessary requirement to the maintenance of continuity in a varnish film. As an element of flexibility, elasticity is related to heat endurance. Hazeltine (147) believes that the degree of oil resistance is a function of elasticity, although the author is of the opinion that this relationship is probably empirical.

ABRASION RESISTANCE. Abrasion resistance of varnishes has definite value from a repair shop or maintenance viewpoint, since accidental impact by tools or cleaning equipment may frequently occur. Abrasion resistance is a measure of the work involved in overcoming effective cohesion of the film and is physically represented as the area beneath the stress-strain curve of the varnish (177).

FLEXIBILITY. Flexibility of a varnish film is a measure of the relative ease with which the films will yield to applied stresses. Although flexibility itself has no immediate significance in the definition of a suitable insulating varnish, it is an important property along with elasticity in defining and determining heat endurance. Hazeltine (132), however, feels that high degrees of flexibility as shown by the mandrel test (heat endurance) are not necessarily indicative of an elastic nature.

BONDING STRENGTH. Bonding strength is a complex combination of tensile strength, plasticity, cohesive strength, and resilience. It is principally a measure of the overall mechanical strength of the varnish required to resist the displacing force of centrifugal motion on conductors held in place by the varnish.

ADHESION. In order that applied coatings may offer the best possible insulation, it is necessary that the film be bound firmly to the apparatus and that it resist all normal forces tending to dislodge it. This basically requires that the varnish react in some small measure with the metal of the apparatus and that this bond be chemically and physically stable under a variety of service conditions.

NON-CORROSIVE TO METALS. Corrosion by varnish films is not a common phenomenon. Few coatings promote continued solution of the base material in a manner usually interpreted as corrosion. As a matter of fact, the green compounds formed by the interaction of copper and the fatty acids of the varnish are almost always of themselves good insulators and usually good corrosion inhibitors. The primary objection to this effect is the reduction in cross-section of fine wires, which may cause or lead to open circuit failure. In relation to corrosion, the two main requirements of a varnish are that the decomposition products attending extended aging do not tarnish adjacent easily-oxidized metals (111) or corrode metals by repeated condensation and evaporation of vapor-liquid systems of lower molecular weight fatty acids, and that the formation of electrolytic products during the aging of the film under the influence of a direct-current potential shall be slight (6). Gardner and Watt (292), for example, cite that most frequent failure in audio transformers results from varnishes decomposing to produce strong electrolytes.

RESISTANCE TO IONIZATION. Varnishes to be used in equipment operating under high potential differences should not favor the formation of corona discharge (249) and should resist attack by continued exposure to the products of corona, that is, ozone, nitrous oxide and nitric acid. Kiernan (250) believes that corona forming in air-filled voids within a treated coil may easily become a hidden danger due to failure of the varnish to resist these discharge products.

ACID RESISTANCE. The use of varnish as insulation about battery installations, particularly concentration cells, requires that it possess resistance to strong acids. It is usual to specify resistance to 10 per cent solution of sulfuric acid

(battery electrolyte), although the test is not strictly service-simulant in that the acid solution may become concentrated through evaporation, thereby increasing the deleterious effects of the acid upon the varnish. Findley and his colleagues (24) believe that acid resistance is not a primary issue since the action of most acids is to further oxidize varnishes.

ALKALI RESISTANCE. Effects of alkali on varnish films may be the result of direct contact with liquids containing high concentrations of ionized hydroxyl groups or the equivalent effect produced by a direct current potential difference existing across a solution of an ionizable salt such as sodium chloride. In either case, the effect is to saponify the oily constituents of the film. Obviously, equipment which operates continually in contact with seawater atmosphere must possess a high degree of alkali resistance.

NON-OXIDIZING EFFECT ON TRANSFORMER OILS. Operation of large oil-filled equipment of expensive design requires that, in addition to complete oil proofness of the varnish applied to various parts thereof, the oil itself shall not be affected by the varnish. Maskell (246) and Norris (307) cite cases of several types of insulating varnish which accelerate the oxidation of transformer oils.

SMOOTH AND GLOSSY FILM. The ability of a varnish to dry to a smooth hard finish reduces the expense of maintenance and eliminates, for example, the danger of particles of metal working their way into the insulation (103).

FILM STABILITY. The varnish film should attain its optimum electrical and mechanical performance qualities within a short time after application and should retain these properties during long periods of exposure to service under a variety of conditions of moisture, temperature and mechanical stress.

DIP-TANK STABILITY. Varnishes subjected to exposure in dip-tanks should exhibit no marked tendencies toward skinning or "livering". Polymerization of baking grades in particular should not progress to the point where the varnish is no longer soluble with the recommended thinners.

HIDING POWER. Where the choice of a varnish is made with the element of hiding in mind, the varnish should possess either high reflectivity, low transmission, or both in sufficient degrees so that thinly-coated portions of the apparatus to be insulated will be adequately "blocked out".

NON-FLAMMABILITY. In any question of transportation, storage, and application of varnish, the element of flammability arises. Limits of flammability are, as a rule, set by the particular use in mind or by specific legislation. Thus, it is possible with some sacrifice in properties, to produce varnishes having flash-points as high as 150 degrees F. The adjustment is a matter of proper selection of solvents, since the solids do not flash under about 600 degrees F.

NON-INJURIOUS SOLVENTS. Varnishes used to impregnate equipment containing enamelled wire, such as relay coils, must not contain solvents which are known to soften wire insulation. Experience has shown that this softening effect is least with aliphatic types of hydrocarbon distillates such as petroleum naphtha or mineral spirits, and greatest with aromatic types such as xylene and toluene.

## TEST METHODS.

### *Preparation of Test Specimens.*

Accurate evaluation of insulation varnish performance necessitates the preparation of suitable films upon which certain operations may be performed with a reasonable degree of ease, as a result of which reliable indications of the characteristics of the varnish may be obtained.

SELECTION OF THE BASE MATERIAL. Considerable variety of practice exists in the selection of the base material upon which the varnish film is cast. Copper, in sheet or rod form, appears to be the most generally used (28), (36), (60), (101), (107), (175), (270), (287), (295). Other metals such as steel (12), (293) find some application. Several workers (13), (166), (172), (247) employ metals easily attacked by laboratory reagents so as to liberate the film in a detached condition. Hazeltine (37) uses wooden blocks in the testing of resistance to weak acids and alkalies.

In many cases (3), (12), (14), (28), (36), (60), (101), (107), (205), (270), (284), (285) paper and calendered cloth are also used, particularly for certain tests, but the author is of the opinion that the selection might be one of convenience rather than based on any strong technical grounds. As a matter of fact, the use of fibrous bases alone, without copper or other metal, cannot be considered service-simulant, and due to variable degrees of impregnation of the varnish into the fibres themselves, this practice does not permit simple interpretation of the results obtained through the use of these specimens.

Selection of the base material should consider first the duplication of actual service conditions as closely as possible, and second, the best fulfilment of the conditions of the test. Metals, in particular copper, meet these requirements for most of the performance tests used in the evaluation of insulating varnish. Evans and Davenport (175), for example, consider that copper metal is instrumental in the strong-acid formation in hot transformer oil, and that the oil resistance test should be conducted with films prepared on copper sheet. However, certain precautions must be taken in interpreting test results obtained using different bases, since, as pointed out by Young and his co-workers (169), (180), different metals activate different thicknesses of film so that a critical thickness must be exceeded before the amorphous characteristics of the film are met. For example, with copper the thickness must be at least 0.3 to 0.4 mil, whereas with galvanized iron the film must be at least 0.8 to 0.9 mil thick, or about 150 per cent greater.

The fact that copper sheets and rods of high purity are commercially available in numerous sizes and finishes at a relatively low cost would preclude the use of almost all other metals in varnish testing.

CLEANING THE COATING SURFACES. It is common knowledge that the surface of the metal to be coated must be thoroughly and painstakingly cleaned. Use is made of various pickling or degreasing solutions in some laboratories to remove the oxide coating of metals and to give the surface a characteristic roughness. The author favors the use of a mixture of about equal parts of soap powder and soap cleanser of fine grit size, applied with #00 steel wool, either using manual or mechanical methods, since this gives a grease-free surface of fairly reproducible roughness. The soap is completely removed by washing, followed by a final rinse in a mixture of equal parts of pure xylene and alcohol.

Coated rod specimens are usually employed for determining acid and alkali resistance (12), (28), (270), (295). Mathes (296) makes use of coated brass rods in a heat cycling salt water resistance test. Stringham (28) utilizes either rods or metal sheets in the testing of water and acid resistance under 100 volts stress, although it is not fully understood in what manner the edges of the sheets are protected to avoid breakdown at thinly-coated portions of the specimens. As is characteristic in all types of immersion tests, the solution or the dialyzed solvent passes through the film under the influence of osmotic pressure differentials (310) and reduces the adhesion of the film. The test is then equivalent to a long-time dielectric strength test at whatever electrochemical conditions prevail at the coated electrodes. In this respect, therefore, the preparation of rod surface is extremely important since if the film is allowed to exercise its normal adhesion, the test measures overall time to first destroy the

adhesion and then to reduce its mechanical and electrical resistance. The practice of highly polishing the rods before coating reduces the adhesion factor considerably and allows the consideration of electrical resistance almost singularly.

That the surface finish of rods is an important element in the resistance testing of films to seawater, for example, is illustrated by the fact that coatings prepared on rods which were polished using #00 emery paper showed proofness of more than 2000 hours, whereas films coated on highly-polished rods repeatedly failed in less than 24 hours. More recent data drawn to the author's attention would indicate that this may be disproportionately greater than the usual case, although repeated tests show the resistance to be favored by adhesive opportunity.

The author knows of no method except that involving the use of a surface analyzer or profile tester which enables the operator to reproduce a given surface finish to the degree of precision apparently required by this test. However, it is felt that highly-polished surfaces, as obtained by buffing, for example, are more easily reproduced by visual inspection than are papered or etched surfaces.

PREPARATION OF SPECIMENS. Since most test results are obtained either from operations conducted on films of definite thickness or from calculations on the basis of a predetermined standard thickness, it is apparent that some accurate method for measurement of thickness is necessary. The most accurate method reported which can be adapted to the measurement of film thickness is probably that described by New (166), wherein a spherometer with buzzer contact attachment is applied to a film under no load and the displacement of the micrometer screw with respect to the base material measured. Using this method, repeated thickness measurements within 0.03 mil are possible. The dial-type micrometer method (20), (177), (285), (295) consists in applying a pressure foot to the film under a dead weight, and has not the disadvantage of the machinist's micrometer of applying torsional stresses to the film, and is desirable in that a constant and measureable pressure is applied to the film. Repeated measurements using the dial-type micrometer can be made to within 0.05 mil, whereas the best precision reported using the machinists micrometer is about 0.1 mil (173). Other methods employ a focus-shift of a microscope (166), and an electronic gage designed for use with non-magnetic bases. The precision of the latter method, as with most instruments of this type, is not better than 0.1 mil as indicated by the data (298), although it is stated that measurements can be made to within about 0.01 mil. It is the author's opinion that the dial-type micrometer best combines the required precision and accuracy with reasonable ease and rapidity of operation, both elements being important to the production-line testing of varnishes.

The problem of selecting the best film thickness appears to be a highly controversial one. Flight (10) and Partz (55) represent a school of thought which advocates using the varnish at the consistency as supplied, whereas others (24), (66), (284), (295) favor coatings of standard thickness. The fact that thicknesses in practice may vary several hundred percent, as pointed out by Schuh and Theurer (177) is not considered as a basis for neglecting control of test specimen thicknesses. Argument in favor of uniform or standard thickness is based on the fact that drying of the film is a function of film thickness, and also that the relation of many properties of films to their thicknesses is not a straight-line function, but approximately logarithmic. Slight differences in thickness particularly in thin films, may result in vastly different properties, even with specimens of an identical sample of varnish.

The necessity of producing test films of uniform thickness has been quite thoroughly reviewed by Ladd (194). Methods have been suggested whereby films may be prepared by spinning (19), (152), spray-coating (152), and the use of a doctor blade (68), (247). Payne (260) has studied the dipping method,

particularly at withdrawal rates of four inches per minute or less, and finds that the thickness is uniform to less than 0.1 mil at very slow rates. The dipping method has certain restrictions, since the initial film uniformity is not necessarily maintained upon re-coating, unless perhaps the withdrawal rates are extremely low. The thickness of second coats is usually greater than that calculated on the basis of first coat measurements. These relationships are shown under Figure 1, where the small crosses indicate

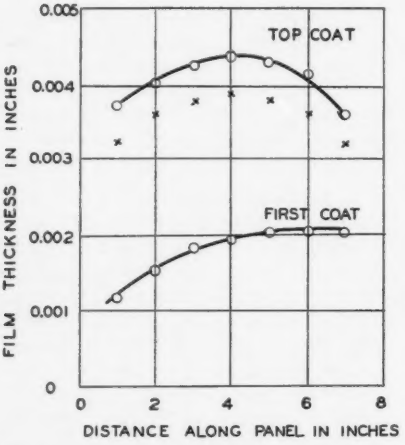



FIGURE 1a.—Effect of Dipping Speeds on Film Thickness. Relationships for withdrawal speeds of about 12 inches per minute. Crosses indicate theoretical second-coat thickness.

the theoretical second-coat thickness. Such variation is probably due to the fact that the base upon which the films have been cast has changed from metal to varnish.

There is no apparent agreement in the literature as to the best specimen preparation temperature or time. Some workers advocate different drying times for different test specimens (50), (55), (270). Others (28), (36), (55), (295) specify that a standard pre-determined drying time shall be adhered to. Findley et al (24) and Partz (55) stress drying of test specimens to equivalent hardness, whereas Thomas and Griffith (247) advocate baking or drying to the point of performance stability. A.S.T.M. uses a preparation schedule which is a multiple of the measured drying time of the film. Obviously, the use of different drying times for different test specimens can become unwieldy. The author has reviewed a government specification requiring as many as 12 different baking temperatures, some only four or five degrees different from each other.

Drying to equivalent hardness, whatever that condition is, may be suitable if that state could be determined with any reasonable degree of accuracy. Also, the

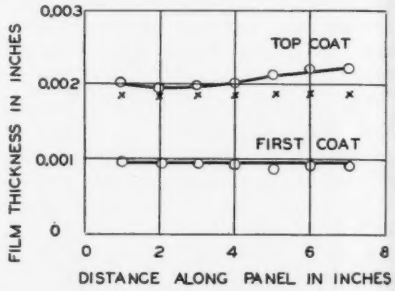



FIGURE 1b.—Effect of Dipping Speeds on Film Thickness. Relationships for withdrawal speeds of 4 inches per minute. Crosses indicate theoretical second-coat thickness.

point of performance stability may be a matter of hundreds of hours baking, which is not only impractical from the testing standpoint, but gives little information regarding the performance of the varnish during the first few days of service. The method of calculating baking or drying periods based on measured drying times merely penalizes rapidly-drying varnishes, since those that dry most rapidly are conditioned but for a fraction of the time that those which dry more slowly are conditioned. For the purpose of testing, preparation times and temperatures should be as nearly representative of the average shop practice as possible, and the preparation schedule should be defined if possible within the limits of the normal working day.

The effects of variable degrees of conditioning have been reviewed by Kienle and Adams (74). Their data on the variation of dielectric strength and power factor with baking time are shown under Figure 2. In these data, the

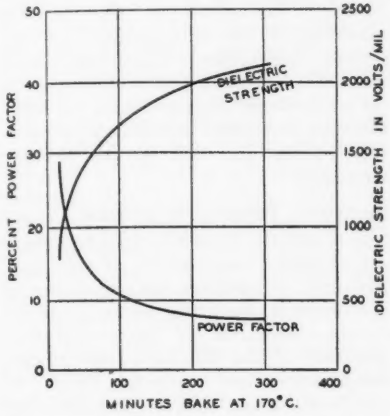



FIGURE 2.—Relationship Between Dielectric Strength, Power Factor, and Baking Time of Varnish Films. (After Kienle and Adams)

measurement of power factor is an excellent means of showing chemical unsaturation and low molecular-weight polymer content of the varnish film due to inadequate or incomplete drying. In cases where accelerated baking information is desired, the work of Kienle (78) shows that if the preparation time is known at two other baking temperatures for which the properties are similar, the time required at a third temperature for similar properties may be computed by use of the following equation:

$$\log_{10} t = \frac{A}{T} - B$$

where t is the baking time, T is the baking temperature, and A and B are constants for the varnish and are evaluated from the data at other temperatures.

The practice of preparing films on one side of copper sheet (295) is highly recommended, since the precision of most tests is considerably improved thereby. For example, in the measurement of dielectric strength of specimens coated on both sides of the sheet, it is usually found that the puncture forms in one film and that the second film breaks down at a point displaced a short distance along the sheet. This is due to the fact that weak spots in both films do not necessarily lie directly opposite each other. Obviously, no accurate estimation of net film thickness can be made at the point of puncture, and hence the dielectric strength cannot be calculated with any reasonable degree of accuracy.

The procedure recommended for the preparation of copper sheet specimens is essentially that described below:

(a) complete cleansing and rinsing of the sheet in grease-proof solvents
(b) measurement of copper thickness at ten-predetermined and symmetrically-spaced points, using a template
(c) backing of two sheets together and sealing at the edges using a suitable masking tape
(d) application of varnish at withdrawal speeds of four inches per minute or less and drying under dust-free conditions with closely-controlled temperature and humidity
(e) separation of the dried sheets and removal of all traces of masking tape adhesive
(f) measurement of total copper and film thicknesses at the same points as in (b), using the template.

### *Control of Test Conditions.*

SPECIMEN PREPARATION. Both the preparation and testing of varnish specimens should be conducted under rigidly-controlled conditions of cleanliness, light, temperature, and humidity, since there is ample evidence of the general effects of these factors on the reproducibility of test results (24), (133), (163), (191).

During the preparation of varnish films, two processes contribute to what is measured as the drying time, namely evaporation of the solvent and the polymerization and oxidation of the non-volatile residue. The rate of evaporation of the solvent is, of course, dependent upon the solvent type, the temperature, and the humidity. Oxidation and polymerization, on the other hand, depend largely on the temperature, the effect of humidity being exerted only during the initial or induction period of drying. Kienle (78) has shown that the baking process follows the mathematical expression for a uni-molecular reaction and that the variation of reaction rate (drying time) with temperature may be expressed as follows:

$$\frac{d}{dT}(\ln k) = \frac{A}{RT^2}.$$

As a rule, the rate of drying increases about two-fold for each ten degrees increase in drying temperature. Thus, slight fluctuations in the drying temperature may produce variable properties of the dried film.

Baking of films may be carried out in forced-draft ovens in which a large percentage of air is re-circulated rapidly over the surface of the specimens, or in a gravity-convection oven in which the temperature differences between inside and outside air create a natural draft. Some laboratories use forced-draft ovens in order to simulate a type of service condition, but the author strongly believes that the variation in properties caused by contamination through dust impingement on the film far outweighs any advantage that this method, as it is practiced generally, might have in the way of service simulation.

The factor mainly responsible for variations in the ventilation of gravity convection ovens is temperature change of the surroundings. Thus, it can be shown that increasing the ambient temperature from 30 to 100 degrees Fahrenheit about an oven operating at 212 degrees decreases the ventilation about 50 percent. Variations in the relative humidity of the surroundings are not usually significant, since the value thereof is normally below 10 percent in the oven. Some attempt should be made, however, to control the amount of moisture in the air, that is, the absolute humidity.

SPECIMEN TESTING. In addition to control of conditions under which specimens are prepared, maintenance of rigid control over atmosphere during testing of specimens is absolutely necessary to the precise evaluation of the performance qualities of the film. For this reason, temperature and humidity must be standardized and closely controlled. The efforts of the American Society for Testing Materials in defining the standard laboratory condition as 77 degrees F. and 50% R. H. have been helpful in this respect.

The necessity for close control in atmospheric conditions is easily illustrated in the heat endurance test, for example. These specimens "breathe" air upon removal and cooling after aging. Specimens which would just crack on flexing after cooling in a dry atmosphere have been shown to be entirely flexible if cooled in a humid atmosphere. The data of Shuh and Theurer (163) given in the table below show the effects of temperature and humidity on the elongation of one-mil films of varnish:

*Percentage Elongation Under Constant Load.*

| *Varnish* | *Relative Humidity at 77 deg. F.* | | | *Temperature at 50% R. H.* | | |
|---|---|---|---|---|---|---|
| | 26% | 50% | 90% | 55°F. | 77°F. | 95°F. |
| A | 1.5 | 4.2 | 49.9 | 0.6 | 4.2 | 54.0 |
| B | 43.0 | 42.3 | 46.6 | 12.5 | 42.3 | 48.9 |

A—10-gallon ester gum varnish
B—Baking enamel

In the testing of resistance to immersion in various solutions, increasing temperatures appear to favor lowered resistance, whereas variations of humidity appear to exert no direct effect.

Electrical measurements are particularly sensitive to changes in temperature and humidity, since in all cases the resistance of varnish films decreases with increasing temperature and moisture content. Curtis (58) points out that the amount of condensed water on the surface of a dielectric increases as a rather large power of the relative humidity. Rylander (138) rates temperature and humidity effects as the major factors in deciding the accuracy and precision of insulation resistance measurements, and shows that the value of resistance may vary 10 to 1000 times as a result of varying atmospheric conditions.

Contamination by dust exerts relatively little effect on the strictly mechanical tests such as hardness, drying time, abrasion resistance and heat endurance. However, electrical tests, and in particular immersion tests under electric stress, are extremely sensitive to contamination by solid particles which may be conductive in themselves, or which are of size sufficient to distort the film.

It has been repeatedly shown that immersion tests on contaminated rod specimens under voltage stress almost always break down at points where some distortion of the film has been effected. This action can best be interpreted by an explanation of the physical property of dust in relation to the film.

Dust may contain both conductive and non-conductive material (24), (133), (300), each of which affects the performance characteristics of the film in a different manner. Conducting particles lower wet dielectric strength and heat endurance, whereas non-conducting particles may lower wet dielectric strength (although not always) and almost always impair the heat endurance. Lint particles, which form a large part of the general type of dust, act as wicks and facilitate the absorption of moisture into the film, reducing the wet dielectric strength. (26), (133)

## ELECTRICAL TESTS.

### *Dielectric Strength.*

The initial dielectric strength of varnished apparatus is often considerably in excess of that required for actual service. This is due to the fact that (a) most varnish formulations inherently insure high dielectric strength, and (b) the initial values of dielectric strength must be relatively high in order that the residual protection will be adequate after prolonged periods of exposure to acids, dirt, and moisture.

The physical mechanism of the dielectric breakdown is as yet not fully explainable on a mathematical basis. The following causes, however, are known to account for failure of the dielectric under applied voltage stresses:

(a) progressive generation of heat due to energy absorption from the applied electric field

(b) ionization of the dielectric material and consequent reduction in resistance with increased energy losses

(c) mechanical disintegration as a result of electrostatic stresses.

TEST METHODS. The short-time (or instantaneous) test is the one most commonly performed. It consists of the application of various voltages to the specimen, starting from zero and increasing to the breakdown value at a uniform rate. The A.S.T.M. test (287) notes, for example, the use of a 2-kva (or larger) transformer supplying an alternating voltage of sinusoidal character between metal discs two inches in diameter in such manner that the voltage increases from zero to breakdown at a rate of about 500 volts per second.

The step-by-step test is designed to lay more emphasis on the time factor, and consists in applying the voltage to the specimen at some fraction of its short-time breakdown strength and increasing this by equal voltage increments at a predetermined time interval.

The dielectric life test, although not commonly conducted, measures the long-time effects of stress and corona on the insulation at some pre-determined voltage, usually higher than the maximum expected in service.

TEST VARIABLES. The dielectric strength test is one to which a high degree of precision cannot usually be assigned. For example, the variations in puncture voltage between adjacent points on the same specimen may be as large as 100 percent. Kasson (57) points out that the difficulties involved in the test make precisions of more than 10 percent unreasonable. The A.S.T.M. procedure (287) recognizes 10 to 15 percent average deviation of the mean as not over-excessive.

The chief factors contributing to variation in test results are as follows:

(a) *Electrode size.* The common presence of weak spots in most applied varnish films accounts for the observed increase in precision with increased electrode size in that the probability of testing a weak spot is large. Counteracting this effect is the fact that using large electrodes may result in the breakdown values being generally higher since the stresses are more uniformly distributed. Apparently in some cases this does not hold entirely true. The author is reminded of certain tests on silicone varnishes which showed 300 percent higher dielectric strength with ¼″ electrodes than with 2″ electrodes and that the standard deviation of tests made with the smaller electrode was considerably smaller.

(b) *Electrode pressure.* Some suggested test methods (55), (270), (284), (295) specify the pressure to be used between electrodes. Rough exploratory tests would indicate that with ordinary pressures (0.3 to 0.5 pounds per square inch) considerable variation in pressure exerts no appreciable effect, whereas pressures in excess of 10 pounds per square inch may result in lower dielectric strength due to distortion of the film.

(c) *Film thickness.* Variations in film thickness exert two independent effects, each of which may act in the same direction. First, dielectric strength is normally lower for homogeneous films of greater thickness, this relationship being considered exponential (287). Second, the extent of drying of a film of varnish, and hence its dielectric strength is inversely proportional to the film thickness.

(d) *Rate of voltage rise.* All materials show characteristic dielectric time-lag curves. That is, for a given wave-crest value and impulse wave shape, the measured dielectric strength varies with the rate of voltage application, being greater with more rapid increases.

(e) *Temperature.* Since the mechanism of dielectric breakdown is partly of a thermal nature, it is to be expected that there will be close variations with temperature. Numerous tests show the breakdown strength of a varnish film to vary only slightly with temperature, particularly at room temperature. Average variations of 35 percent, based on the lower temperature values, have been observed for changes as great as 100 degrees C., which would indicate that even five-degree changes at room temperature are more or less insignificant.

(f) *Moisture content.* Varnish coatings which have absorbed their full capacity of moisture may show reduction in dielectric strength of as great as 75 percent. Errors due to surface leakage and flash-over are more frequent the greater the moisture content of the film. In this respect, then, the humidity of the specimen conditioning room should be maintained as nearly constant as practicable. Some loss in moisture content is affected as the voltage stress is increased, but the rate of voltage application is usually too rapid to make such losses of more than academic significance.

(g) *Frequency and wave-form.* The dielectric losses in varnish films under alternating current stress are composed

of ohmic and hysteresis losses (38). Variations in frequency (at constant temperature) produce hysteresis losses according to the following equation:

$$E = k/f^n$$

where $k = K(a + (1\text{-}a)/T^{1/4})$

$f$ = frequency

$T$ = duration of voltage stress

and where for varnish films the nominal values of the constants are:

$K = 1.75$

$a = 0.675$

$n = 0.137$

Since dielectric breakdown of varnish is predominantly thermal in nature, and in view of the fact that the peaked waves of the same maximum voltage as flat-topped waves will produce less heating effects within the film, it is to be expected that variations in wave-form will exert some effect on the breakdown voltage. Some test methods (107), (287), (295) specify an allowable deviation from a pure sine wave-form and, therefore, fix the distortion within certain limits. Morgen (202) advises that the supply voltage should be reasonably free from harmonics, and that the "insulation under test is a capacity load, and if the capacitance of the specimen is resonant with the inductance of the transformer and supply equipment at the frequency of any harmonics that may exist in the supply, the result due to this harmonic will be a very great increase in the current, and the resultant voltage wave-form will be affected".

(h) *Transformer capacity.* As the impressed voltage is raised across the insulation, ionization in the dielectric permits a definite current to flow. The size of this current depends on the capacity of the transformer. Thus, a film which will support an ignition-coil spark without puncturing (10,000 volts or more) may show breakdown at 500 volts using a 5-kva transformer. The author has found as little as 2-kva difference in ratings between transformers to affect the breakdown values for voltage significantly.

Certain other factors enter into a discussion of the test variables of dielectric strength measurements, these factors being less easily interpreted than those foregoing. For example, the inhomogeneity of the film is responsible for considerable variation in the dielectric breakdown test results, perhaps more so than any other factors mentioned. This inhomogeneity may be inherent in the film (such as non-uniformly distributed gel structures) or may be introduced during the preparation of the test specimen (dirt and debris). Holmes (89) has advanced an equation expressing the mean voltage breakdown of a material as a function of its inhomogeneity:

$$E = \bar{E}\left(1 - S\,(\text{erf}^{-1}\frac{N\text{-}1}{2N})\right)$$

where

$E$ = mean voltage breakdown of a set of samples

$\bar{E}$ = mean voltage breakdown of a unit sample

$S$ = inhomogeneity constant

$N$ = total number of samples tested (may also be taken as the electrode area in cases where the thickness is small)

$\text{erf}^{-1}$ = read "quantity whose error function is", and is equivalent to the quantity found from the probability relationships.

SIGNIFICANCE. Considerable difference in opinion exists regarding the value of the dielectric strength test. Findlay and his associates (24) believe that the wet dielectric strength test is second in importance to the heat endurance test. Summers and Ross (274) consider the test particularly useful not only in defining quality of raw materials, but in selecting substitutes. Arnold (139), on the other hand, believes that the dielectric strength test should be ignored in view of the fact that thickness required for adequate mechanical strength automatically insures high dielectric strength.

The dielectric strength test, particularly after water immersion, is important from the standpoint of service. Conditions have been repeatedly observed which favor the condensation of moisture on windings and which must, there-

fore, have some effect on the dielectric breakdown values. Thompson and Mathes (297), for example, report electrical equipment operating in regions having 15 feet of rainfall in thirty days during two months of the year, and where the average relative humidity is 92 percent and over for 24 hours of the day during four summer months. In cases of this kind, the wet dielectric strength test is an appreciation test, indicating the necessity of an engineering estimate of safety factor in the design of insulation to meet this condition.

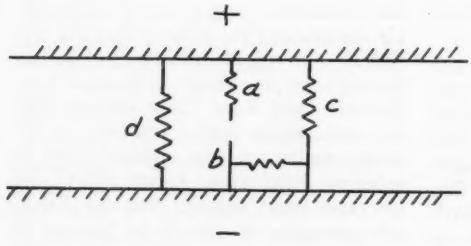
In addition to actual design requirements, experience has shown that varnishes which exhibit satisfactory performance in service also possess a level of dielectric strength below which a varnish may be suspected of being inferior in quality. In this respect, dielectric strength tests can be looked upon as an index of quality.

Examination of dielectric strength test results in the light of other properties often gives valuable information about the varnish in general. For example, low wet dielectric strength, good heat endurance, and excessive drainage would indicate insufficient bodying of the varnish during the cook, whereas low wet dielectric strength, low heat endurance, and low drainage might indicate contamination of the film.

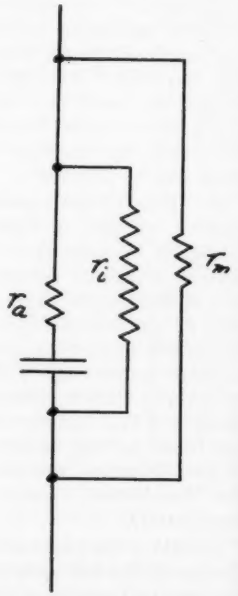
*Insulation Resistance.*

Insulation resistance is a measure of the current developed through or across a dielectric surface due to ionic conduction under an impressed voltage difference. Resistance is, therefore, classified into two types—surface leakage and volume resistance, each of which exerts its influence quite independently of the other. Thus, materials which have high surface resistance may not show exceptionally high volume resistance. Since low surface leakage is a property of special insulating materials and not particularly a requirement of insulating varnish in general, the discussions herein will be limited to volume resistance.

TEST METHODS. Insulation resistance is a complex quantity in that the dielectric material acts as a condenser as well as a resistor. Peek (73) illustrates this property according to the following figure:




SCHEMATIC DIAGRAM




CIRCUIT DIAGRAM

The resistance of the substance itself, $r_i$, is an intrinsic property and is very large. The resistance, a, is a partially conducting element and is shown as a resistance in series with a capacitance as $r_a$. Elements c and d are partially-conducting fibers which extend across the gap and are shown as $r_m$. With direct current voltages, the only measureable flow of current exists across $r_i$ and $r_m$. With alternating currents, as distinct from direct, the current is a vector sum of the currents existing through all three resistances and the capacitance, and is accordingly much greater, the effective resistance being, therefore, lower.

Measurement of insulation resistance is commonly made by one or more of the following methods, Wheatstone bridge, "megger" instruments, direct deflection galvanometer, and series voltmeter. The general features of each instrument are as follows:

(a) *Wheatstone bridge.* This method is an application of the common practice of balancing unknown resistances against a known resistance, using a sensitive galvanometer or electrometer capable of detecting very small currents.

(b) *Megger instruments.* Use of these instruments is made where the actual resistance is not sought with more than casual accuracy, but where assurance is wanted that the resistance lies above some minimum value. It is also useful in following resistance changes with other parameters, as for example during the baking or drying cycle of the varnish applied to certain primary insulation. These instruments are either of the constant potential or of the variable potential types, and may be obtained in a variety of voltage ratings.

(c) *Direct deflection galvanometers.* This procedure is most applicable to high resistance measurement and for that reason is most widely used. A potential is applied to the insulation and the deflection of a sensitive galvanometer noted. This deflection is compared to that obtained using a standard resistance. Due to absorption, a certain time is required for the insulator to become charged, and a comparable time required for dissipation of this charge through the insulator and thence through the galvanometer. The A.S.T.M. standard method (289) specifies a charging and discharging time of one minute. Knowing the various shunt resistances and constants for the instrument, calculation of resistance is relatively simple.

(d) *Series voltmeter.* This method is slightly more accurate than the first two methods outlined above, but requires an outside source of voltage. The procedure is to take comparative readings of the line voltage and the differential voltages when the resistance is connected in series with the meter. The relationship of the various voltages becomes:

$$R_1 = R\frac{E - E_1}{E_1}$$

where

$R_1$ = insulation resistance
$R$ = voltmeter resistance
$E$ = line voltage
$E_1$ = voltmeter reading during test

TEST VARIABLES. Perhaps the most serious objection to the use of the Wheatstone bridge is that its useful range is confined to the measurement of resistances less than one megohm, primarily because of the low voltage used. It is believed that this method might be useful in conjunction with an electrometer if standards of resistance were available which remained fairly constant in value and which obeyed Ohm's law. Van den Akken and Webb (151) suggest a type of bridge for measurement of resistances of the order of a thousand megohms consisting of a modification of Maxwell's commutator bridge, wherein a small capacitor is charged and discharged about 30 times per second and accordingly taken as equivalent to a high resistance of the value, $1/(nC)$. This forms one arm of a Wheatstone bridge, the other arms being two resistances and a resistance shunted by a capacitance.

The main objection to the "megger" instrument is the unrealiability of resistance readings at high values. These instruments have a logarithmic scale but,

although the scale divisions at the infinity end of the scale are severely cramped, most instruments of this type will measure five megohms. Older instruments are not usually of the constant-potential type; that is, the voltage impressed varies directly with the speed of the generator. Obviously, speed must be calibrated in terms of the voltage gradient desired during the test. At any rate, the instrument is practical for commercial testing only.

The upper limit of measurement with the direct deflection method of test is about $10^7$ megohms, using a galvanometer of maximum sensitivity and a voltage gradient of about 180 volts. Instruments operating at higher voltages are not recommended in view of the relatively low dielectric strength of some varnish films, particularly when wet. For, instance, the electronic device described by Bushman (258), while entirely suitable in its range (5 to 20,000 megohms) impresses 500 volts across the specimen, and, therefore, could not be used to measure the insulation resistance of varnish films less than about four mils in thickness.

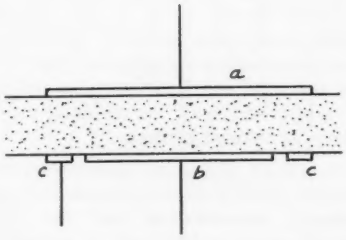
The general test variables are temperature, moisture, voltage, surface leakage, and electrode contact or area, each of which affects the resistance measurement in some characteristic manner:

(a) Resistance of insulation is known to change considerably with small changes in temperature. In general, due to poor thermal conductivity of varnish films, some time is required for the establishment of uniform temperature throughout the insulation. Measurements should be made under very closely controlled conditions of temperature, that is, about one degree C., plus or minus.

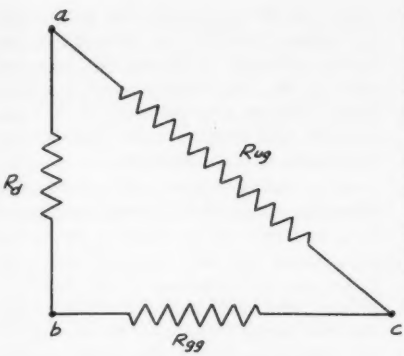
(b) Relatively dense materials, such as varnishes, may require days to assume equilibrium with a given ambient humidity, but often lose moisture rapidly when subjected to an atmosphere of lower water vapor tension. For this reason, specimens which have been immersed in water for a period prior to resistance measurement should be tested in an atmosphere of controlled humidity so that the loss of moisture during the measurement will be fairly constant from specimen to specimen. Pre-conditioning at high humidity should, if possible, be followed by measurement of resistance at the same humidity.

(c) The fact that Ohm's Law does not appear to hold at values of high resistance makes it necessary that the voltage applied be specified and be held rather closely to a prescribed value. This is particularly important where the voltage gradient approaches the dielectric breakdown value for the specimen, since ionic conductance of the insulation increases and allows a large current to flow.

(d) Surface leakage is an important element of any insulation resistance measurement. As such, it may be either measured directly and corrected for, or the circuit may be so designed that the leakage current does not flow through the galvanometer. Guarding the test electrode is probably the easiest way of taking into consideration surface leakages. For example, in the case of the direct deflection tests employing one guarded electrode, Greenfield (251) considers it necessary to maintain the potential difference between guard and the unguarded electrode at somewhat higher value than between guarded and unguarded electrodes to insure collection of all surface leakage. Diagrammatically, the circuit is as follows:




ELECTRODE ASSEMBLY




GRAPHICAL REPRESENTATION

where

$R_d$ = direct resistance (volume)

$R_{ug}$ = resistance between guard and unguarded electrode (volume plus surface resist.)

$R_{gg}$ = resistance between guarded and guard electrode (surface resistance)

Miner (226) points out that the necessity for considering surface leakage is eliminated by maintaining the guarded and guard electrodes at substantially the same potential difference, therefore reducing the current flow between these elements practically to zero.

(e) Comparisons of resistances of films of the same thickness presupposes maintenance of a constant electrode area. Perhaps the only practical method of accomplishing this is to use metal foil electrodes and to apply them to the film by means of a high-resistance adhesive, such as petroleum jelly. The electrode must be in intimate contact with the film at all points unless some method be used to correct for area not in contact. This problem of contact is an especially significant one in the measurement of wet insulation resistance or of resistance at high temperatures, since volatilization of moisture or gas creates blisters between the electrode and the film, reducing the contact area. In temperature-resistance measurements this "gassing" is accompanied by inflection points in the resistance curves.

SIGNIFICANCE OF TESTS. The usual large drop in insulation resistance of varnish films after water immersion (in some cases more than 90 percent) illustrates the possibilities of the test in determining the effects of water on varnish insulation. The precision of measurements on wet insulation is, however, extremely poor, so that often more than four parallel determinations are required to show effects conclusively. The test is, therefore, perhaps only a qualitative one. Mathes (296) considers that the test serves only the purpose of indicating other properties of the film, and is of the opinion that the magnitude of the values for resistance has no significance in varnish testing.

Tests conducted by the author show that the initial resistance data can be repeated with a fair amount of precision. In this respect, the test may be used as an indicator of quality and as a basis of comparison between several materials. Varnishes showing differences in resistance, however, cannot be conclusively suspected of being dissimilar. By the same token, any attempt to identify a specific formulation by means of insulation resistance data is, for all practical purposes, impossible.

### *Power Factor.*

Power factor is that fraction of energy lost during the regular charging and discharging of a dielectric in an alternating current voltage field. It is defined as the cosine of the phase angle between voltage and resultant current, which angle depends on the ratio of resistance to capacitance in the dielectric material.

Insulating varnishes, as a rule, are not required to meet power factor specifications, although their power factors are

generally low because of their characteristic formulation. In view of this, there are no test methods at present which outline a detailed procedure to be used in the measurement of power factor on insulating varnishes.

The selection of the test equipment is largely a matter of personal choice and convenience, although the most popular type of apparatus from the standpoint of sensitivity and ruggedness is the Schering bridge operating at 60 cycles. Housman (252) describes a bridge having a range of capacitance of from one half the value of a standard air capacitor to about 20,000 times the standard, as well as an automatic recording bridge useful in life tests on insulation. Hill (259) describes a meter designed to measure the quantity, Q, defined as the ratio of capacitive reactance to resistance. For large phase angles, Q is large and approximately equal to the reciprocal of the power factor. The circuit employs an oscillator as a voltage source at any desired frequency acting across a variable capacitance, to which may be coupled the test capacitor.

TEST VARIABLES. The factors influencing the test results in power factor measurements are temperature, frequency, moisture content, leakage, and electrode contact, each of which is important for the following reasons:

(a) *Temperature.* Aside from the fact that losses in dielectrics are due to the existence of ions, temperature changes result in variation of the randomness of the orientation of polar molecules, and hence produce changes in the degree of dielectric losses.

(b) *Frequency.* For the electrical insulating materials in the range of frequencies usually met with in electrical instruments, the power factor nearly always decreases in value with increasing frequency, the extent of this change being proportional to the actual loss. This is to be expected, since at some very high frequency the natural period of oscillation of the polarized units would be so small that compared to the frequency of measurements, it would be of no influence.

(c) *Moisture content.* The power factor value is affected by moisture in two ways: moisture increases the apparent polarity of the insulation as a whole, thus affecting the material as a capacitance, and moisture also reduces the resistance of the insulation.

(d) *Leakage.* The total capacitance between two parallel electrodes separated by a dielectric is the sum of the actual capacitance of the material, the edge losses due to fringeing of the flux lines in air from one electrode to the other, and the stray losses due to passage of the flux lines from the high-potential electrode to ground by way of air space. A judicious choice of shielding (for stray losses) and guarding (for edge losses) reduces the magnitude of these errors considerably, and is absolutely necessary for accurate work. Extended calculations are, of course, possible for the correction of these errors, but these have been worked out for only simple electrode shapes (288).

(e) *Electrode contact.* As pointed out by Hoch (53), the presence of an air-gap of only 0.001 inch in series with a specimen having a dielectric constant of 2.5 will have the same effect as increasing the specimen thickness by 0.0025 inch as far as measurement of capacitance is concerned. Therefore, absolutely intimate contact with the film is necessary for each electrode. Electrodes are best applied to varnish films in the same manner as discussed under the section on insulation resistance measurements, that is, using petroleum jelly or similar substitute.

The power factor test should be reproducible to at least 5 percent, but probably to not more than an absolute value of 0.0001, depending on the care exercised in the preparation of the test specimen and the errors inherent in the equipment.

SIGNIFICANCE OF TESTS. The power factor test is generally recognized as a very sensitive indication of the presence of impurities in insulation, although the work of Caldwell and Payne (288) would indicate that the changes are primarily due to changes in resistance rather than capacitance.

For any given varnish formulation power factor of the dried specimens is an invaluable indication of the purity of the component parts, as well as the extent of the cooking or bodying operation. Housman (181) advocated the use of the test as an indication of manufacturing variations as well as of quality. Kienle and Adams (74) have followed the degree of baking of varnish films using power factor as an index of the completeness of oxidation and polymerization, as is shown under Figure 2. Freed (267) notes that low power factor shows possibility of good performance or life during field service of the insulation.

*Arc Resistance.*

Arc resistance is a measure of the ability of a surface to form an insulating barrier to current flow under the destructive influence of an intermittent or continuous arc. The A.S.T.M. method (286), at present the only standard test method, employs a 12,000-volt arc produced by a 1.2-kva transformer acting across two spaced tungsten-tipped steel electrodes. The arc is intermittent during the early stages of the test and gradually increases in frequency of application until continuous. This intermittent operation of the arc is resorted to in an attempt to spread the breakdowns of various types of specimens and thus afford a measure of classification based on arc resistance.

The author has followed the procedure of coating films on scratch-free, clean, glass panels and baking or drying for a pre-determined time. Glass is perhaps the most convenient base to use which possesses high arc resistance (200 to 250 seconds).

TEST VARIABLES. The test is perhaps the least precise of all electrical measurements. There are some factors, however, which appear to influence the test results on varnish films, such as electrode weight, arc positioning, specimen cleanliness, external drafts, and moisture content of the film. A discussion of these factors follows:

(a) *Electrode weight.* Excessive weight on the film at the point of contact with the electrodes has the effect of embedding the electrode tips in the film and thus contributing toward premature breakdown through tracking. Sufficient weight should be employed so that only the minimum pressure is acting in the direction of maintaining full electrode contact.

(b) *Arc positioning.* The heat produced by arcing results in a reduction of resistance in the air directly above the arc, and the tendency of the arc is to travel in an upward arc over the specimen. This has the effect of drawing the arc away from the surface of the specimen. Tests conducted using glass tubes of semi-circular cross-section, under and through which the arc is confined to travel, show that the variability of the measurement can be reduced, since the heat of the arc is confined to the surface of the specimen. The main objection to this procedure at the present seems to be that the glass tube had a tendency to darken due to condensed decomposition products in the arc, thus obscuring the end-point. Exploratory tests on inverted arcs (electrodes on the under side of the specimen) show this procedure to hold some merit.

(c) *Specimen cleanliness.* Obviously, any material present on the surface of the test specimen which may contribute to the development of a low resistance path between electrodes will reduce the arc resistance of the specimen. Such material may be grease, dirt, or conducting liquids formed by the leaching out of water-soluble electrolytes by condensed moisture.

(d) *Draft.* In addition to disturbing the arc, drafts tend to cool off the region about the electrodes and effectively increase the apparent arc resistance of the material.

(e) *Moisture content.* Although varnish films containing considerable moisture suffer reduction in arc resistance, it is doubtful if slight changes in moisture content can be detected by this test.

SIGNIFICANCE OF THE TEST. Arc resistance is not generally a service-simulating test, since, with the exception of direct current motor commutation, intermittent or continuous arcing across insulation is a rare condition. The test does, however, show some indication of the thermal stability of the varnish since the test is essentially a measure of the resistance to carbonization. Using the A.S.T.M. test procedure baking varnishes show arc resistances of about 170 to 190 seconds, and air-drying varnishes of about 100 to 140 seconds. Water immersion for any considerable length of time prior to the test may show reduction in these values of up to 50 percent.

NOTE: The remainder of this paper will be published in the August 1949 issue.

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# PART TWO—REPRINTED ARTICLES

The articles contained in this Part Two represent those which, in the opinion of the editorial staff, are the most valuable which have been gleaned from recent issues of contemporary publications.

This material was formerly published under the heading "NOTES". It has been given somewhat more prominence and made more readable to enhance its value. The editors' purpose here is to provide a convenient repository to readers of the JOURNAL, for articles originally published elsewhere which contribute to the purpose of the Society; to further the advancement of naval engineering.

# NOTES ON THE BEHAVIOR OF H.M. SHIPS DURING THE WAR.

ACKNOWLEDGMENT.

This article by N. G. HOLT, R.C.N.C. and CAPT. (E) F. E. CLEMITSON, R. N., is reprinted from the Transactions of the Institute of Marine Engineers, 1948 Vol. LX. The paper was read and discussed at a joint meeting of the Institution of Naval Architects and the Institute of Marine Engineers.

This paper is intended to give a general picture of technical difficulties which arose during operation of the fleets in the last war. These difficulties were increased by the widespread nature of operations in all kinds of sea conditions and climates varying from the tropics to well within the arctic circle, and by the many radical changes in fighting conditions brought about by the advent of modern weapons.

The improvements in aircraft in the twenty years between the two wars had increased their menace to ships to such an extent that the provision of adequate counter measures for protection was no easy problem.

Much increased H.A. armament with improved fire control systems, and the introduction of radar and W/T techniques, all had to be accommodated in ships, the majority of which had been designed ten to twenty-five years before. Such changes necessitated serious inroads into stability and freeboard and equally serious reductions in the space available for accommodation. Accommodation difficulties were thus doubly accentuated by the additional personnel required to man the new equipment and the reduced space available.

Despite such handicaps and the age of many of them, it can be said that H.M. ships withstood the severe tests of war very well. Although certain weaknesses were revealed, which had not been brought to light under the less rigorous conditions of peacetime, on the whole both the ships themselves and their machinery gave proof of their sound design, and, even though operational demands severely curtailed opportunities for proper maintenance, were able to keep at sea and fulfil their functions to an extent somewhat above reasonable expectations. The fact that certain ships were able to remain at sea for as long as three months, with no opportunities for maintenance at all, speaks for itself.

In general, although of course there were plenty of maintenance difficulties and comparatively minor troubles, no real weaknesses were revealed in the case of the machinery, despite the wear and tear to which it was subjected, though certain new troubles arose in the case of boilers, e.g., "scab pitting" on the insides of tubes (occasioned by prolonged steaming and difficulties in maintaining adequate purity of feed-water), circumferential fatigue cracking of tubes in certain small vessels, brought about by hard steaming and rapid changes of speed (remedied by improving circulation), and the accumulation of hard bonded deposits on the outside of boiler tubes due to the presence of salt water in the oil fuel, which also led to rapid

deterioration of furnace brickwork. Apart from these there were no abnormal troubles of a general nature.

Many valuable lessons were learned regarding the vulnerability of machinery, particularly in connection with the effects of shock due to underwater explosions and "near miss" bombs. The extent of damage to machinery from shock was greatly reduced by various remedies, evolved early in the war, including the replacement of cast-iron castings by less brittle materials where possible, and the provision of shock-absorbing mountings, etc.

War experience also led to considerable improvements in methods and appliances for fire fighting, and before the end of the war our ships were well equipped to deal with serious fires.

The nature and length of the war, with ships closed down for action periods persisting over many hours and even days, gave ample evidence of the necessity for improved ventilation and lagging and the control of wild heat in machinery spaces. Much attention is still being given to these aspects by the Admiralty Departments. Improvements in equipment and in the technique of its application are essential if a heavy price in space and weight is not to be paid for improved habitability; experience clearly shows that to hold the scales evenly in time of peace between military equipment and amenities for personnel is a difficult problem, and yet one which must be continuously and correctly solved over many years if our ships are to be in the best fighting trim and ready for any emergency.

*Introduction.*

During the second World War ships of the Royal Navy saw long and exacting service in every part of the globe. It is certain that they were never more tried and strenuously fought in the whole course of the Navy's history. Operations were so widespread that rest for ships, machinery and men, for repair or recuperation, was hard to provide, and many ships endured long and dangerous service with continuous steaming for weeks on end over a period of years. The ships were to a great extent planned and designed before the outbreak of war, in some instances years before the outbreak of war, but in general they proved adequate to meet conditions of service more exacting than had ever been contemplated, or that could have been accepted under peace-time conditions.

One remembers personally a passage from the Mediterranean in 1943 in a "C" class cruiser, nearly thirty years old, which was returning home to be paid off for good. This ship had been sailing the Indian Ocean for about three years, during which time she had steamed many thousands of miles without any real period in dockyard hands for refit. To be a technical officer on the staff of various commanders-in-chief during those critical years was a most interesting and valuable experience as well as a great privilege and responsibility.

The conditions of service in operations which varied from the Arctic to the Pacific did inevitably reveal some deficiencies, and it is the purpose here to indicate the type of difficulty which arose. It will not be possible to say definitely whether these should or could have been foreseen nor perhaps in every case will it be possible to indicate the remedy even now. All these technical lessons have received close attention and such experience will undoubtedly benefit future construction. It is hoped by this self-criticism over points of detail that the great achievements of H.M. ships throughout the war will not be obscured.

## Bad Weather Structural Troubles.

### *General.*

Structural failures due to bad weather are unlikely to be met with in peacetime, because prudent seamanship is then exercised, but in war operational necessity may be overriding. Steaming speeds even in bad weather are relatively high. Hence it is inevitable that weaknesses should be uncovered and, although the great majority were superficial, a number were significant and perhaps worthy of mention.

One very bad gale experienced by the Home Fleet in the spring of 1943 just south of Iceland had one or two interesting aspects. The wind (speaking from memory) increased rather quickly from force 7 to 8 to a hurricane logged at force 12 and the resulting sea was very high and relatively very short. H.M.S. *King George V* and other ships, including an escort carrier and destroyers, were practically hove to at 6-7 knots with the wind and sea forward of the port beam, and there is no doubt that the destroyers were making better weather of it than the much larger battleship. The light displacement hulls of the destroyers, provided speed can be adjusted to suit the sea conditions, are always happier than the heavy battleship. In this case H.M.S. *King George V* shipped a good deal of water, including one particularly bad sea which carried a well-secured cutter clean over the side from the catapult deck, shifted and heavily damaged boats on the boat deck about 30 feet above water-line, and caused a fair amount of flooding to "B" engine-room via the ventilation trunk. It might be mentioned here that the *King George V* class suffered appreciably on service from the relative lack of sheer forward. This sheer had been reduced below what was desirable to meet the requirements for the fire of the main armament right ahead and the freeboard forward had later been reduced still further by the unavoidable increase of displacement caused by the application of war experience after the completion of the ship. It is perhaps only fair to add that the shipping of a bad sea on this particular occasion was not affected by this lack of sheer forward.

In this same storm, but a few miles away from the rest of the fleet, H.M.S. *Sheffield* lost the roof of one of her 6-inch turrets when the turret was struck by a heavy sea.

This particular storm is well remembered personally because of the argument which prevailed in the ward-room for some days afterwards on the actual height of the waves on this occasion. One school of thought was convinced the waves reached a height of 80 feet whilst the other half of the wardroom willingly agreed to 45 feet but would not go higher; the captain himself was firmly in favor of 80 feet. Having occasion on arrival in Iceland to visit some destroyers and the escort carrier to deal with minor repairs the opportunity was taken to collect opinions from the commanding officers (all experienced seamen) on this burning question, and it is interesting to note that in each case the commanding officer quoted a height of wave roughly corresponding to the height of his own eye above water, i.e. 45-50 feet in destroyers and the escort carrier and 80 feet in H.M.S. *King George V*. It was not possible to measure wave heights or lengths during the passage but the most conservative estimate gave a height of 40-45 feet and a length of 550-600 feet.

An experience of H.M.S. *Nelson* when passing through a gale in the Bay of Biscay in the summer of 1943 gave an interesting comparison with H.M.S. *King George V*. The latter ship had a period of about 14 seconds and the former one of about 11 seconds, and this very quick period in H.M.S. *Nelson* was quite surprising in so large a ship and resulted in a large proportion of the men's kit lockers lying flat on their backs throughout the gale.

Some of the county class cruisers endured possibly the worst weather conditions in their service over a period of years in the Denmark Strait. In many ways they were ideal for the work, being large and comfortable ships with splendid accommodation and large freeboard. However they were the first of the Treaty cruisers, fifteen to twenty years old; they had been lightly built originally so as to comply with the Treaty, and war-time requirements had added considerably to their displacement. The result was that the general structural strength was near the limit, and in some ships general leakiness was a continual trouble.

In H.M.S. *London,* for example, war-time additions included hangars, extra side armour and increased H.A. armament. It was appreciated that the increased load would require structural compensation so the sheer strakes had been doubled over a considerable length. This action, however, raised the neutral axis of the girder and as a result the stresses in the shell became disproportionately high. These showed themselves by leaks in oil fuel and reserve feed tanks, at the glands, etc. These defects were made good by fitting an oil fuel settling and sullage system and by fitting external butt straps to the outer bottom below the bilge keels. The original arrangement of shell butts in this class, which had been perfectly satisfactory for many years, provided for internal straps only which were stopped short at the boundary angles of the longitudinal bulkheads, a device perhaps more attractive as a weight saver than as an arrangement for standing really heavy service.

### *Weather Decks.*

Weather decks gave a certain amount of trouble due mainly to hard driving in bad weather, and also in some instances due to over-light scantlings and construction. Destroyer forecastle decks had appreciable areas of 5-7 lb. plating with single riveted laps, and serious leaks developed when these vessels were driven into a head sea. Such leaks, not in themselves structurally serious, had a most serious effect on habitability below the deck where a large part of the men's living space was located. A satisfactory solution, without appreciable increase in weight, was achieved by changing to all-welded forecastle decks in later new construction. In destroyers then under construction the plating was changed to a minimum thickness of 7 lb. with double-riveted laps.

### *Breakwaters.*

Breakwaters on forecastle decks also gave trouble in destroyers and in some of the larger ships. In destroyers, the shape of the breakwater was changed to a form of cusp and, wherever practicable, all brackets were worked on the after side. In general, all brackets such as these should have the toe of the bracket ending on a girder and, particularly if riveted, should have a small lug or flange worked across the toe to avoid concentration of stress at the toe of the bracket.

### *Pillaring.*

Pillars under forecastle decks in both cruisers and destroyers required strengthening against weather damage from the shipping of heavy seas.

### *Superstructures.*

In the storm referred to above one destroyer unexpectedly shipped a sea which distorted the fore end of the after superstructure. A door in the fore end was driven in and caused considerable flooding of the officers' cabin flat below. This door had been provided for convenient access to a warhead store, but it was subsequently blanked. This and other experiences demonstrated that square endings to forward ends of deck houses were bad, and that weather deck doors and hatches should be kept to a minimum, carefully sited to get the best possible shelter from the sea and designed with deep coamings and strongbacks if highly inconvenient distortion and leakage was to be avoided.

### *Bilge Keels.*

Bilge keels gave a certain amount of trouble which, in one or two instances, rendered the ship non-operational until repairs had been effected. In a few cases of destroyers and smaller vessels the bilge keel was peeled away completely, starting at the forward end, and in one other case, even more awkward, the bilge keel was left hanging, giving the ship an unknown draught and the captain considerable anxiety. Panting stresses causing local fractures of the hull plating at the forward end of the bilge keel sometimes occurred. In practically every case, the trouble could be traced to two main causes, namely (1) carrying the bilge keel too far forward, and (2) giving the bilge keel too abrupt an ending. In general, and particularly in smaller high-speed ships, there is everything to be lost and little to be gained by carrying the bilge keel far forward of amidships. In destroyers, for example, the average length of bilge keel forward of amidships is about 17 per cent of the water-line length and 10 per cent would probably be a safer figure for hard driving in bad weather. In the steam gunboats the percentage as designed was 19 per cent, and this was reduced to 6 per cent as a result of damage during their first sea trials.

### *Carrier Flight Decks.*

Fleet carriers suffered fairly severe structural damage under the forward corners of the flight deck on more than one occasion due to pitching into bad head seas. The conflicting requirements of seaworthiness forward and a large square ended flight deck for good flying qualities were very difficult to reconcile, and as a result of war experience the latest carriers have been modified forward to give a more gradual flare out from the water-line.

Escort carriers suffered more frequently than fleet carriers from damaged structure forward owing to bad weather. Experience shows, not unnaturally, that the open exposed forward end of the flight deck in the escort carrier was much more vulnerable than the plated in but heavily flared bow of the fleet carrier. Also, while the welding of the main hull in the escort carriers was generally sound, the welding of the superimposed structure supporting the flight deck was often very bad and accentuated the effects of bad weather damage.

### *Barbettes.*

A structural defect which was exposed in bad weather in a number of battleships, both old and new, was severe straining and fracture of the rivets in the heavy angles connecting the barbette ring armour to the deck plating. It has to be remembered that this barbette armour, about 40 feet in diameter, comprises, so to speak, a very rigid cylindrical island in a sea formed by a large area of flexible weather deck plating and to keep two such items effectively united, when the ship is straining heavily in bad weather is a difficult problem. A

lot of sea water found its way below, causing considerable inconvenience and loss of habitability. This was cured by the fitting of flexible sheet steel curtain plates between the top of the barbette angle and the barbette. Since the war, the variation of stress in the weather deck across a typical section through a large barbette opening has been investigated at the Naval Construction Research Establishment and as a result better structural arrangements could now be made in such cases.

## Rudders and Steering Gear.

With the enormous number of miles steamed at high speed and the practice of zig-zagging there is no doubt that the rudders and steering gears of our warships led an extraordinarily hard life. A general policy of stiffening the plating on the fore end of the rudders of many types of ship was found necessary as fatigue cracks developed owing to vibration and perhaps some degree of corrosion fatigue, and portions of plating were sometimes found missing or peeled back on docking.

One case quite different from the above but also of considerable importance was that of the rudders of *King George V* class. In these ships, for speed and stability reasons, the sections in way of the rudder were of pronounced "Y" form with distinctly concave sections to the floor plates in the vicinity. This ship is such as to prevent the shell plating from giving appreciable resistance to sideways thrust on the lower pintle. The floor plates were connected to the shell by single angle bars single riveted on each flange, and considerable stretch and shear of these rivets occurred within the first two years of service. (Much the same thing can be seen in the screw fastenings aft in racing yachts where the weight of the lead keel acts as the agent straining the fine follow after sections.) Serious leakage occurred into the steering compartment and the sideways movement of the top of the rudder head became so appreciable that fracture of the stern casting became an imminent risk. It was essential that entirely adequate modifications to this structure should be made before sending these ships out to participate in a possibly long-term Pacific war, so double angles were worked connecting the floor plates and shell, and additional athwartship webs were introduced to support the stern casting at the rudder gland.

It is perhaps a little unusual in modern times to find a ship losing her rudder at sea, and it becomes even a little astonishing when that ship has two rudders. However, this actually did happen to an American-built *Captain* class frigate where the weight of each rudder was taken by a nut screwed on the lower end of each rudder post. These nuts were secured by small locking plates welded to the nut and to the rudder structure in each case. Making the small assumption that the rudders were lost one at a time and not simultaneously, advocates of twin rudder steering will be interested to learn that the commanding officer had not apparently been aware of any lost steering efficiency prior to the loss of both.

Late in 1944 one of the aircraft carriers of the *Illustrious* class reported difficulties with steering when steaming at high speeds. There was excessive time lag, after a turn, before the rudder followed the wheel back to midships position, and with the wheel amidships there was a tendency for the rudder to wander about 2½ deg. to port. Also it was virtually impossible to steady the ship if port rudder was required at speeds over 25 knots.

The duties of an aircraft carrier necessitate implicit reliance on the ship's steering capabilities at high speed, so that, with these ships getting ready for operations against Japan in the British Pacific Fleet, the reports were a matter for serious concern.

Much investigation was carried out, and trials in another ship of the same class indicated similar tendency, although not so severe, or such as to cause undue

anxiety. The remaining ships of the class, however, did not appear to have noticed any such difficulties.

In the design stage of these three-shaft ships it was known that the rudder was overbalanced with the ship going ahead and the centre screw working, but trials had shown that the steering gear—of the electro-hydraulic type—always had sufficient margin of capacity to deal with the maximum torque ahead or astern although the designed pump relief valve settings had had to be increased. At the same time peak torques occurred near the mid-position of the rudder, so that the pumps were working at high loads for the greater part of their running time.

The investigations in the case of ship concerned brought out the following facts. There was wear in the steering pumps, no doubt from the considerable running time at high loads, which seemed sufficient to cause the wander of the rudder near the mid-position which was being experienced. The holding torque at about 10 deg. port rudder was considrably greater than originally. This seemed to point to an increase in the overbalance of the rudder—a new rudder, although to the same design, had been fitted since the ship was first built, and it was reported that the troubles had only been noticed since this new rudder was fitted. This overbalance would also increase the tendency of the steering pumps to wear.

The remedies which were adopted at the time were in the nature of palliatives. New or refitted steering pumps were installed, the relief valves were reset to a higher pressure, and a limit imposed on the maximum revolutions of the centre shaft. Later—although this did not take place until hostilities had ended—a new and modified rudder was fitted in which the area forward of the axis was reduced in order to reduce the over-balance effect.

All these remedies seem to have cured the trouble, but it cannot be said that the precise cause of it has yet emerged. On the whole it probably lies in the overbalance of the rudder which resulted in the steering pumps working at high loads for the greater part of their running time. Consequently, there was cumulative wear in the pumps, and after a certain period they became incapable of exerting sufficient capacity. It is significant that the ship in which the real trouble was experienced was the one of the class which had carried out the most steaming, and the two ships which appeared to have no trouble had steamed the least.

### Bridge Structures and Masts and Gun Blast.

There has been an increased requirement in modern warships for space in the bridge structure both for vital operational equipment and for sleeping accommodation for key personnel. These bridge structures have, in consequence, assumed quite massive proportions in cruisers and battleships and have caused serious windage problems in many cases.

In *King George V* class, the tall cliff-like front and sides gave good screening to the forward part of the compass platform but made conditions in the admiral's bridge below extremely draughty and unsatisfactory for operations in cold climates. A joint effort by the staffs of the Director of Naval Construction and the National Physical Laboratory including the testing of a screening draught of air from a special fan, failed to provide a satisfactory solution.

In H. M. S. *Nelson*, conditions on the bridge frequently most uncomfortable due to fumes from the funnel, particularly when the ship was steaming head to wind. In this case, the top of the funnel was appreciably below the top of the bridge structure, and it is surprising that energetic action had not been taken some years before the war to cure so obvious and troublesome a defect.

Some of the windage problems on open

bridges in cruisers were greatly accentuated by additional equipment fitted just before the bridge front. This equipment interfered markedly with the vertical air flow which is so important a factor in protecting exposed bridge personnel. The close range weapons and zareba mounted on the roof of "B" main turrets in *County* class cruisers were particularly troublesome in this respect.

Masts in warships of all types were severely tested during the war years. The loading and windage on the masts to meet the increasing requirements for wireless and radar were quite beyond anticipation when the ships were designed, and in smaller ships, i.e. cruisers and destroyers, vibration trouble was sometimes serious. Such trouble was unnecessarily accentuated in the case of tripod masts by inadequate cross staying, this merely consisting in most cases of a triangle of horizontal stays half-way between the base and the top of the tripod. With a height of mast of 40 feet or more, it was not difficult for a pair of struts, when giving way and vibrating to the increased load at the top of the mast, to sway at mid height like the hips of a Hawaian dancer. This was easy to cure by the introduction of one or two additional stays.

To meet the increased loading on destroyers' masts, without increase of weight, it was decided to change to a braced lattice structure, and this type of mast has proved satisfactory.

On the whole, however, warship tripod masts fully justified their design. One case was particularly interesting and significant. This was H.M.S. *Duke of York's* mainmast which, in the fight with *Scharnhorst,* had a good half section of both mainmast and port strut shot away and yet carried its considerable load through very bad weather for several days while steaming back to harbor.

Weather decks subject to blast from heavy guns fired at low elevations showed considerable vertical violent movement in areas between the main bulkheads. Damage to equipment in sick bays, dental surgeries, ventilation trunking and electrical cable leads from weather deck movement often caused considerable inconvenience. Minor bulkheads should therefore be stopped about 2-inch short of such weather deck areas and, in general, equipment should not be secured to such bulkheads but supported from the deck below. Alternatively, any equipment such as ventilation trunking, cable trays, etc., which must be supported by hanging straps from the weather deck should have a good spring system incorporated in such supports.

## Main Machinery.

### *Impulse Blading Fatigue Failures.*

During the first three months of the war a disconcerting number of failures occurred in the first row ahead impulse blades of the new *Tribal* class destroyers which had the effect of putting ships out of action for repairs at a time when they were urgently needed.

Investigation showed that the breakages were due to fatigue of the material, caused, presumably by the combined effects of vibration in resonance with impulses from the nozzle vanes, and the repeated shocks inevitable with partial admission. The trouble was cured by machining out the rotor groove and fitting ⅞-inch wide blades of a specially strong section and reduced height, in place of the original ¾-inch wide blades.

Later failures of the same type occurred in one or two other destroyers and one aircraft carrier. The steam bending stresses in the high-pressure ahead Curtis wheel blades of a number of ships have been calculated, and the results show that the maximum stress (which usually occurs when a small group of nozzles is in use alone) is a satisfactory criterion of the safety of

such blades. The *Tribals*, with their high proportion of failures, have considerably higher stress than other ships where failures evidently occurred only in borderline cases, probably due to some defect in workmanship.

The calculations also revealed that under certain conditions the impulse blades in some aircraft carriers then nearing completion would be in danger, and instructions were accordingly issued that the smaller nozzle groups in these ships were not to be used alone.

Practice now is to limit the steam bending stress to one-half the lowest figures known to have caused failure.

## Boilers and Associated Fittings.

### *Scab Pitting.*

In the early stages of the war what seemed to be a new type of internal corrosion of boiler tubes made its appearance. It took the form of circles of hard oxide crust (hence the name "scab pitting") under which pitting rapidly developed, and was found mainly in the fire-row tubes, those nearest the furnace suffering severely and being attacked on the fire side of the tubes towards the lower ends. Other fire-rows were similarly affected to a lesser degree, with occasional attacks in the outer rows of tubes.

As hostilities continued, a large number of ships were affected, and had to be laid off for retubing of varying extent; in some cases tubes required renewal after only two years' service. The tubes and furnaces of cylindrical boilers have also been attacked by scab pitting.

It seems quite certain that the trouble was due to salt contamination of the boiler water which war conditions rendered much more likely to occur than peace-time steaming. Corrosion of this nature is an electro-chemical phenomenon, and is initiated at places where access of oxygen to the metal surfaces is non-uniform. Space forbids a detailed description of the mechanism of the attack; but briefly, the products of the consequent reaction are not soluble except in the presence of sodium chloride, and are therefore immediately precipitated on the surface of the metal, causing a blanketing effect and stifling further action. When, however, sodium chloride is present, the products, which are ferrous chloride and sodium hydroxide, are soluble, and the ferrous hydroxide formed by the resultant interaction is precipitated some little distance away, settling on the metal as a thin membraneous disk which acts as a semi-porous diaphragm, allowing the free passage of ions, but preventing the diffusion of oxygen. What is known as an "oxygen concentration cell" is set up, and anodic attack of the metal begins, and proceeds with increasing intensity. The longer the process continues the thicker the incrustation or "scab" becomes, and the more efficiently does it act as a diaphragm. Underneath the scab, therefore, a pit develops which deepens with great rapidity.

Areas which are subject to more rapid heat interchange are more liable as areas of potential partial breakdown —as was found in practice. Once the scab formation has taken place pitting will develop irrespective of the condition of the boiler water, i.e. even if salt contamination is eliminated. This fact, combined with the rapidity of the attack, provided the serious feature of the trouble. Moreover, if was found impracticable to remove the corrosion products from the pitting—the only way of halting the progress of the corrosion —by means of power-driven boiler tube cleaning brushes. Even when tubes have been removed and cut up for examination, cleaning by pickling has proved difficult. The answer seems to be in the use of chemical treatment for the feed water, i.e. boiler compound, which has now been generally adopted in the Navy.

*Circulation Troubles.*

In the early stages of the war a number of sloops, and later minesweepers, began to experience trouble from severe distortion of fire-row tubes, accompanied by circumferential cracking which necessitated tube renewals and the consequent immobilization of the ships at times when they were badly needed.

The failures appeared to be due to overheating, and since they occurred most consistently in ships having auxiliary machinery of the reciprocating type, and boilers that were relatively lightly forced in comparison with the larger boilers fitted in destroyers and other major warships, it was at first suspected that overheating was attributable to contamination of the feed water with oil derived from the auxiliary exhaust system, as a result of excessive use of oil for internal lubrication of auxiliary engines. Metallurgical examination of tubes which failed in this way, often after very brief periods in service revealed that the cracks were transgranular and characteristic of fatigue. It was found possible to produce similar cracking under laboratory conditions in tubes subjected to alternate heating and quenching by passing a douche of water through the tube, after a relatively small number of cycles. These experiments showed that the failures were attributable to intermittent or irregular circulation, and theoretical investigation of the circulation characteristics of the boilers showed that the circulation margin was very much less than was at first supposed.

In all cases the boilers were of very squat design, having a low natural circulation head and shallow tube banks. The lack of flexibility of the tube nests was also a contributory factor. A cure was found in fitting unheated downcomer tubes and altering the curvature of the fire-row tubes to provide increased flexibility.

The effect of fitting perforated plate baffles in the water drums of the boilers was also tried with a view to improving the circulation in the fire-row tubes by increasing the turbulence in the water drums, and thus preventing the separation of any steam which might be generated in the outer row of tubes.

In later designs the height of the boilers and depth of the tube banks was increased and the arrangement and curvature of the tubes was modified so that the slope of the tubes at exit from the water drum was as near vertical as possible.

Modern methods of boiler design can give a more accurate assessment of circulation characteristics than was previously the case, but this is a point which still has to be watched. It is customary now to carry out circulation trials to determine the safe forcing rate, using the criterion that with boiler pressures of the 400 lb. per. sq. in. the depression ratio should not be greater than 0.5.

*Bonded Deposits.*

Late in 1942 several ships—battleships and cruisers—of the Home Fleet started to report trouble with excessive external deposits on superheater tubes and portions of the generator tubes. These deposits were of a hard type, bonded to the tubes, and proved incapable of removal by ordinary cleaning methods. As a result, considerable increases of air pressure were necessary when steaming—one ship reported an increase from 8 to 11 inch at full power. Steaming efficiency was thus reduced, and ships had to be laid off for more frequent external boiler cleaning. Subsequently, similar troubles occurred in ships operating in various parts of the world.

This problem has been dealt with at length in a paper recently presented to the Institute of Marine Engineers. It will suffice to say here that the conclusion which has been reached is that these bonded deposits are due to emulsification of the fuel oil with contaminating seawater, and that their formation can only be prevented by ensuring that the oil reaching the burners is free from seawater contamination.

The type of oil supplied to the fleet up to 1942 did not emulsify when brought into contact with sea-water. If there was water present in the oil, it separated readily and could be discharged overboard; or if it did reach the burners they usually became extinguished immediately, and no bad effects resulted.

In 1942, however, supply considerations and the course of the war forced the use of cracked fuels from the U. S. and Caribbean areas in the Home Fleet and certain other stations. This type of fuel readily emulsifies when brought into contact with seawater, and it is possible to burn oil containing even up to 10 per cent of water without much evidence at the burners.

At the same time the ships concerned had occasion either to displace oil by water, or to ballast the fuel tanks, and in many cases trouble was experienced with leaky tanks. Also, it is fairly certain that a considerable amount of wet oil was supplied to ships during the war period.

It is thus easy to see that there was every likelihood of seawater contamination taking place in certain classes of ship, and it is worthy of note that no great evidence of trouble with bonded deposits has come from destroyers and smaller ships which do not displace oil by water, or, normally, ballast fuel tanks.

Another point of interest, and indeed one which may provide food for argument, is than Venezuelan fuel contains vanadium, and at first it was thought that the vanadium content was the cause of the trouble. This theory has, however, been rejected in favor of the sea-water contamination theory, although quite possibly vanadium does play some part in the process.

### *Refractories.*

Various troubles were experienced during the war with furnace brickwork. They were not in themselves of major importance, but they often had a considerable effect in reducing the availability of ships and increasing the already heavy load on maintenance and repair staffs.

The chief defects which occurred were spalling and erosion of bricks, together with breaking and cracking, and burning of brickbolts. As might be expected, the throat bricks lining combustion tubes suffered most.

There were various causes. The material, and to some extent the design of the bricks and brickbolts, were not sufficiently good to stand up to the higher temperatures and forcing rates which were by then being employed in naval boilers. The relatively small amounts of peace-time steaming, mostly at low powers, did not really show this up, and it was left to the prolonged steaming and high powers of war-time operation to do so.

The burning of inferior fuels had its effect also, and the contamination of the fuel by sea-water, which has been referred to above, was a very significant factor in causing slagging of the brickwork.

Temperatures and forcing rates are continuing to increase, and with these increases it is becoming essential to improve both the materials and design of furnace refractories and brickbolts, a fact which is fully recognized, and which is receiving the attention it merits.

## Gearing, Shafting and Propellers.

### *Illustrious Class—Interaction of Propellers.*

The aircraft carriers of the *Illustrious* class, three-shaft vessels, have suffered from a longitudinal critical speed in the centre shaft which occurs slightly below full power. The fact that the centre propeller works behind hull structure causes an unusually large cyclic variation of thrust, which is further accentuated

when the vessel is turning, because the slip stream of one wing propeller then impinges on half the area of the centre screw.

When running at the critical speed the three per revolution impulses from the three-bladed propeller coincide with the natural frequency of the system, and the applied force is multiplied some eight times by resonance. This is sufficient to increase the alternating force when turning to ±187 tons, a figure considerably greater than the steady thrust.

This large force leads to a considerable fore and aft motion of the main gearwheel, and the resulting rapid sliding of the tooth surfaces has been the cause of rapid wear in the flexible couplings between the turbines and pinions, more pronounced in some ships of the class than in others.

Other defects attributed to the excessive vibration have been loosening of rivets in the main thrust block seat, wear and loosening of the centre tube bush, and difficulty in keeping the stern gland tight.

These troubles had their effect in keeping the ships out of action while repairs were carried out. In particular, the loosening of the centre stern tube bush made its appearance at an awkward time when the ships were being prepared for Pacific operations.

In H.M.S. *Illustrious* the experiment has recently been tried of fitting a five-bladed propeller to the centre shaft, which appears to have cured the trouble completely. It has a doubly beneficial effect, because it moves the critical speed to a lower power where the forces are much reduced, and also the five-bladed propeller has an inherently smaller cyclic variation than one with three blades. The amplitude of vibration has been reduced to about one-quarter of the original figures, and the motion of the main thrust block is now barely discernible, even when turning.

### *Wear-down of A-frame Bushes.*

A problem which arose in Eastern waters, and to some extent in the British Pacific Fleet, was the very rapid wear-down of lignum vitae A-frame bushes in certain ships, mainly cruisers, which necessitated laying off the ships for unduly frequent dockings on a station where docks were few and far between, and ships often had to steam long distances to reach them.

The cause of this trouble, of which the U. S. Navy also had some experience, is still not entirely clear. One likely cause was poor quality lignum vitae which had also been stored under unsatisfactory conditions; but local conditions, such as the scrubbing action of coral in suspension in the sea-water, or possibly the action of other marine organisms, may have been contributory.

What ever the true causes, the results were not without their effects on operational requirements.

### Arrangement of Machinery.

In some cases action damage revealed certain weaknesses in the layout of the main machinery, in that it showed that unduly serious loss of steaming capabilities could result from a single hit, in a particular position, with consequently serious effects on the safety and fighting efficiency of the ship.

One particular example is the loss of H.M.S. *Ark Royal*. Like the aircraft carriers of the *Illustrious* class, H.M.S. *Ark Royal* was a three-shaft ship with the boiler-rooms arranged athwartships, and the engine-rooms similarly, abaft the boiler-rooms. This made, in fact, a very neat and convenient arrangement

of machinery, but it meant having a large fan flat extending the full width of the ship above the boiler-rooms, and similar spaces to carry the uptakes.

The single torpedo hit from which the ship subsequently sank was abreast the starboard boiler-room, which immediately flooded. The flooding quickly spread, via the fan flat and the uptake spaces, to the centre boiler room, and then, although more slowly, to the port boiler-room by the same means. In consequence, all steam supply was lost, and this was the biggest factor in the eventual loss of the ship.

After that it was essential to take measures to remedy this deficiency as far as possible in the ships of the *Illustrious* class. It would have been too extensive a modification to have made the uptakes watertight and the main immediate measure adopted was to increase the height of the dwarf bulkheads, over the line of the boiler-room longitudinal bulkheads, both in the fan flat and in the uptake spaces.

A later modification was made which separated the port uptake trunking from the others entirely, to a height well above the hangar deck. A watertight bulkhead was erected between this new trunking and the double trunking as a further safeguard. The air downtakes were also subdivided into separate leads to the port, centre, and starboard fans by watertight trunking.

Later in the war H.M.S. *Indomitable* was hit by a torpedo abreast the port boiler-room. Some serious flooding again resulted, but the dwarf bulkheads proved their worth in sufficiently restricting it to enable steam supply to be maintained in the other boiler-rooms.

The arrangement of the machinery spaces in H.M.S. *Warspite* was also not ideal from the viewpoint of vulnerability as the six boiler rooms were concentrated forward and all steam and power was lost when the ship was struck by a radio controlled bomb off Salerno. This layout, however, was imposed by limitations of the reconstruction immediately prior to the war which involved the acceptance of these arrangements.

### *Failure of Ventilation Supply in Machinery Compartments.*

Another weakness which action damage in certain ships revealed was connected with the failure of electric power —consequent on damage outside the compartments—in one or more main engine-rooms. This led to stopping of the supply and exhaust ventilation fans, all of which were electrically driven at that time, with the result that the compartments very soon became untenable, although the main machinery was undamaged.

The answer to this was to fit at least one steam-driven exhaust fan in main engine-rooms, and in important auxiliary machinery compartments—except those containing electric generators where arrangements were made for the existing motor-driven ventilation fans to take a supply direct from the generators. Thus, provided steam is available to run the machinery (or, in dynamo rooms, the dynamo remains running) it is now possible to keep conditions in the compartments sufficiently good to enable the personnel to remain, and keep the machinery operating, at least at low power.

### Damage To Propeller Shafting.

One or two cases of torpedo damage in the vicinity of inboard propeller shafting resulted, in addition to flooding, in some distortion of the shaft which, nevertheless, kept turning. In so doing it succeeded in tearing away adjacent watertight bulkhead glands from the bulkheads and thus considerably increased the area of flooding.

In order to obviate this possibility de-

signs of flexible bulkhead glands were evolved which allowed some degree of distortion of this shaft or of the bulkhead without causing any damage to the gland. After some initial troubles these designs have proved successful, and are now generally fitted in larger ships in positions where failure of the glands might cause the risk of serious flooding.

It was also considered that somewhat similar risk could result from the actual fracture of a propeller shaft inboard, in that the tendency would be for the broken outboard end to withdraw aft, and the shaft coupling flanges to rupture bulkheads in the way. In fact, it is believed this actually happened in one U. S. cruiser.

Some consideration has therefore been given to the possibility of fitting trailing collars to the propeller shafts at the aftermost plummer bearing, in order to prevent the shaft moving aft in the event of fracture.

Yet another innovation which war conditions showed the necessity for was gear capable of locking a damaged shaft—which could not be trailed—at as high a ship speed as could be provided by the remaining shafts without undue vibration. The old type of shaft brake was incapable of holding the shaft at speeds of more than about 8 knots, and the new type locking gears have been fitted from the later stages of the war.

It is unfortunately, still necessary to slow the ship right down before these locking gears can be engaged, and the obvious requirement—but a difficult one to meet—is for a type of gear which can be engaged at any speeds or, alternatively, for a shaft disconnecting coupling which can be readily disengaged.

## Shock Damage.

This important question has been fully dealt with in a paper by Com'r.(E) A. D. Bonny, R.N. The effects of shock damage were so far-reaching, however, that some brief remarks on it in this paper will not be out of place.

Before the late war began the problem of damage to machinery due to shock had not been treated as being one of importance. The machinery was in general robust, as it had to be for naval service, and had withstood, generally satisfactorily, the vibration met with at high speeds, and the shocks due to firing of guns and dropping of depth charges.

Late in 1939, however, H.M.S. *Belfast* was mined by a magnetic mine on the sea bed, and the very extensive damage to machinery and the hull structure caused by this mine explosion some 80 to 90 feet away from the hull focused attention on this new type of attack. Many more incidents of a similar nature followed, and the various types of enemy non-contact mines, as well as the explosion under water of near-miss bombs, caused a considerable number of ships to be laid up for long periods of repair.

In most cases the machinery and electrical equipment suffered greater damage than the hull itself. The greatest part of the damage to machinery equipment was caused by the fracture of cast-iron castings, although considerable subsidiary damage and derangement resulted from the tripping of governors, defects in electrical gear, priming of boilers in some cases, throwing around of loosely secured gear, and so on.

It was very necessary to investigate the fundamentals of this problem and find remedies, or at least palliatives, quickly, and much intensive work was done which is still going on. The obvious remedies in the case of the machinery were to re-design the machinery suitably to withstand shock, and to replace cast iron by a more ductile material—cast or fabricated steel—but wartime production difficulties precluded such steps being full implemented.

The immediate steps, then, were to modify the existing machinery in ships to minimize the risk of damage from shock. This was done in various ways, many of them of makeshift character—

alteration of existing chocks and the addition of extra chocks, often of wood, under the main mass, additional supports for overhanging masses, provision of stretching lengths on holdingdown bolts, alterations to the retaining keeps of turbine feet to make them yield under the shock forces anticipated, etc., etc. Where general design considerations made it impossible to make the machines in themselves fully shock-proof, resilient, i.e. flexible, mountings were designed for fitting between machines and seatings.

Non-contact explosions and near misses were found to cause a tendency to trip the governors on turbo-driven auxiliaries, notably turbo-generators (Diesel generators were also affected), with the awkward consequences of a total or partial loss of electric power and other services at critical moments.

As an immediate remedy gags were fitted to the trip gear, and instructions issued that they were to be left in whenever ships were at sea. The possibility that, on a sudden emergency, the watchkeeper might be unable to operate the over-speed trip was accepted and, in fact, no instance of damage from this cause can be recalled.

Designs of trip gear with sufficient reentrant angle to resist the degree of shock to be expected were also put in hand, and towards the end of the war fitting of these commenced.

## Fire Fighting Arrangements.

The advent of the war brought an almost immediate realization of the inadequacy of the equipment then borne in H.M. ships for coping with fire hazards.

The first reactions were to increase fire main pressures, and the numbers of fire hydrants and isolating valves in fire mains, and to re-design branch pipes and nozzles to produce either a jet or a spray, in order to make them more suitable for both smoke driving and smothering. The numbers of portable fire extinguishers of the chemical type carried were also increased, and improved designs produced. In this respect particular attention was paid to the hangars and flight decks of aircraft carriers, to guard them against the severe fire risk existing with aircraft and petrol supply systems.

As the war went on, however, and experience was gained, numerous other innovations were introduced, some of the more important of which are mentioned below. It may be noted, incidentally, that, although considerable use was made of chemical fire extinguishers for certain requirements, in general, water in the form of a fine spray, or steam, proved the best means of extinguishing fires.

### *Steam Smothering.*

The use of steam as a smothering agent had been applied to boiler-rooms before the war, in view of the possibility of oil fuel fires. During the war it was extended to main engine-rooms, and eventually to almost all machinery compartments.

### *Foam Equipment.*

A number of improvements were effected in the design of continuous foam generators, making them simpler and easier to operate. The knapsack foam outfit is of some interest. This consists of a foam compound tank suitable for carrying on the operator's back, and a branch pipe and nozzle with a suction connection for coupling to any convenient fire-main connection, the whole making

a readily portable apparatus which can be effectively used in most positions in a ship.

Foam tubes were also fitted in boiler-rooms and certain other machinery spaces, in conjunction with foam generators outside the compartment, for dealing with liquid fuel fires where foam might be a more suitable smothering agent than water or steam.

*Equipment of Aircraft Carriers.*

Apart from increased fire fighting appliances such as have already been mentioned, and also such gear as mobile $CO_2$ fire extinguishing units for flight decks and hangars (U.S.N. water fog nozzles were also tried out), much attention was paid to the problem of dealing with the risk resulting from rupture of petrol tanks combined with flooding and the spreading of large quantities of petrol vapour through portions of the ship.

Instructions were issued concerning the careful isolation of the affected parts, particularly the breaking of all electrical circuits, and air-driven portable pumps were supplied for dealing with flood-water, and air-driven portable fans for clearing petrol vapour. Also petrol vapour detectors were developed for use both under damage conditions and as a routine measure in aircraft hangars.

*General.*

Much attention was paid to the general reduction of fire and smoke risks in ships by the removal of as much inflammable material and fittings as possible, fire proofing of wood, and such measures as the introduction—as far as production considerations would permit—of steel furniture. In all these measures, of course, there had to be some degree of compromise, more particularly under tropical and arctic conditions, on account of habitability considerations.

Finally, in order to ensure that all officers and men understood the value of this new equipment and the correct method of using it, fire-fighting schools, where much invaluable training was given, were instituted at all main naval bases, both at home and abroad.

A full description of the present state of fire-fighting arrangements in H.M. ships is to be found in a paper by Mr. P. G. Haywood to the Institution of Engineers and Shipbuilders in Scotland, 1947.

PUMPING, FLOODING, OIL FUEL, AND PETROL INSTALLATIONS.

Seven years of war brought about an increase in the number and capacity of fixed pumps, but on the whole the layout and arrangement of the suction systems remained relatively unchanged, although the main line suction system fitted in cruisers and carriers was subjected to criticism from the fleet from time to time.

Experience soon showed, however, that the necessity for a truly portable pump of adequate capacity was paramount for dealing with flooding in places inaccessible to the main suction line, or where the line itself was damaged or choked.

Early in the war improved types of portable electric pumps were introduced in H. M. ships to replace the old type submersible pumps, and were supplied on a progressively increasing allowance basis.

Emergency Diesel generators were fitted in ships which either had none or

in which the existing Diesel generator capacity was too small, in order to supply essential electrical circuits should all steam power have failed. Later, as the essential electrical requirements were so many, Diesel-driven emergency pumps were introduced.

Finally, fixed internal suction stand pipes were fitted in ships, for use with the portable pumps and in order to facilitate their operation.

### *Aviation Fuel and Petrol Stowage.*

Mention has already been made of the measures introduced in aircraft carriers to deal with the risk of damage to bulk petrol stowage, and the spreading of quantities of petrol vapour about the ship.

It is a very grave risk, as some of the U. S. aircraft carriers found to their cost, but the Royal Navy were fortunate during the war in having little experience of damage to bulk petrol stowage.

Apart from aircraft carriers and other ships operating aircraft which—cruisers and the like—became progressively fewer as the war went on, every effort was made to dispense with bulk petrol stowage, or else to stow the petrol in readily jettisonable upper deck tanks. In this endeavor the introduction of Diesel-driven ships' boats and emergency pumps helped considerably.

The standard arrangement for carrying aviation fuel or petrol in bulk was to stow it in separate cylindrical tanks placed in compartments which were kept flooded with sea water. This method proved very successful in avoiding major fires and explosions due to rupturing of the tanks as a result of damage in the vicinity, although in the case of aircraft carriers it had the disadvantage of restricting stowage capacity by comparison with the practice of stowing aviation fuel in double bottom tanks forming in effect part of the ship's structure.

In fact, no case of a tank rupturing due to action damage occurred, and in only one case—H.M.S. *Liverpool* in 1940 did a serious fire or explosion result from damage to the distribution system.

### *Oil Fuel—Difficulties experienced due to Degradation.*

In a previous section of these remarks some mention has been made of the troubles experienced with bonded deposits in the gas passages of boilers, and with refractories which have been—at least in considerable measure—put down to contamination of the oil fuel by salt water.

Other troubles resulted from this cause, too, and it may be of interest to examine the history very briefly.

Pre-war Admiralty fuel oil did not emulsify with water—this emulsification is the root cause of all the troubles—the adventitious water was readily eliminated by settling. During the war, however, a number of changes took place in the Admiralty specification for boiler fuel oil, all of which were made with the object of widening the sources of supply and facilitating to some extent the supply of the large quantities of fuel oil required, more particularly from American sources.

These changes were, however, attended with difficulties, chief of them being the increasing tendency for the fuel oil to emulsify with sea water. The introduction, namely in the United States, from 1935 onwards—and increasingly during the war—of deep thermal and catalytic cracking processes, chiefly in order to attain the necessary large production of aviation fuel, led to the degradation of the residues used as a basis for the production of fuel oils. The problems of

high viscosity, high specific gravity, emulsibility, and sludging which are present when using boiler fuel oils containing residues which have been deeply cracked, have become of primary importance with the introduction of water ballasting and water displacement systems. For example, war time and post-war Admiralty fuel oil emulsifies with water readily up to theoretical water contents of 74 per cent. These emulsions are very stable, of high consistency, and are not broken by simple heat treatment. Very hard emulsions containing large quantities of water are of such high consistency that they cannot be pumped.

War-time conditions made for increased liabilty of water contamination. Incomplete removal of the ballast water owing to the rapid turn round of tankers at loading points in the U.S.A. resulted in wet cargoes arriving in Great Britain. In cruisers and smaller ships ballasting of oil tanks resulted in the contamination of the oil remaining in the tanks, and this was not removed by the pumps. In the modern battleships, water displacement of the oil has the same effect, while leaks caused by enemy action or bad weather were an additional factor in all cases. Tankers, however, have been largely immune from this latter trouble owing to the positive outward oil pressure when their tanks are full.

The troubles due to emulsification were as follows:—

(*a*) Wet cargoes arrived at oil fuel depots, and turned these into mixing stations where dry cargoes were blended with wet ones in an attempt to keep within the 0.5 per cent Admiralty specification for water. This, however, was impossible, and the limit was raised to 1 per cent which was, even so, frequently exceeded.

(*b*) The emulsion collected in ships' tanks and was often increased by further ingress of water. During refits the contaminated oil and sludge had to be removed and became a serious embarassment to the shipyards which had limited storage capacity and no outlet for the waste material.

(*c*) It was found impossible to remove very badly contaminated sludge by pumping and it had to be removed by hand labor.

(*d*) The salt present in the water caused the troubles with boiler refractories and choking of gas passages, already discussed.

The size of the problem can be realized when it was estimated that the intake of contaminated oil at the port of Liverpool was of the order of 10,000 tons per annum. There was no ready means of recovering it, and a large proportion had to be taken out to sea and dumped.

The increasing use of heavy cracking in refinery practice means that the problem will continue and, indeed, increase. A great deal of work is being carried out to find methods of dealing with it, and some promising results have been obtained. The crux of it is to keep the water out of the oil.

## Habitability and Amenities.

Making due allowance for the changes in operating conditions which are inevitable in any new war, it is still fair to state that a high proportion of the troubles experienced in habitability would have been found out and probably largely rectified had sufficient regular long and thoroughly realistic exercising been carried out in the twenty years of peace between the two wars. A financial policy which provides for ships in commission but not with the oil fuel to keep them at sea makes but a poor contribution to preparedness, so that although the Director of Naval Construction and Engineer-in-Chief made the utmost use of their opportunities, sea experience comparable to the varied conditions in which ships served in this war was not obtained.

Ventilation was always a very serious problem in operating ships under war conditions. Ventilation trunking had been made very thin—often unduly so in view of its location—and, with mushroom tops in exposed positions on weather decks, was vulnerable to flooding and gun blast. Probably a retrograde step was taken when the old screw down type of mushroom was discarded about thirty years ago due to bent screw shafts and general lack of maintenance; a better policy would have lain in holding to the principle of this type of mushroom and improving the design.

War conditions ruthlessly exposed the gross inefficiency of mushroom top inlets fitted on the weather deck forward and larger inlets or outlets located in higher and more sheltered positions to meet requirements for a group of compartments are now favored. This, of course, leads to other difficulties in maintaining adequate headroom under large sectioned ventilation trunks running fore and aft; it is also not ideal for isolation of ventilation systems in the event of action damage. Moreover, if much discomfort and loss of efficiency, particularly under tropical conditions, is to be avoided, it is most desirable that ventilation should be kept running in all practicable periods for compartments which are manned when a ship is closed or partially closed down for action continuously for some hours; the grouping of ventilation systems may lead to more centralized control of ventilation, perhaps by means of a number of master switches in damage control headquarters. Tropical conditions and atomic explosions are both arguments in favor of such control even for compartments which are air conditioned.

Even when the weather conditions or location of inlets made it practicable to run the ventilation system fully it was not unusual in northern operations to find a mess deck of 60 men supplied by 30 punkah louvres, of which one only would be half open and the rest firmly shut off against the arctic air. Bad sleeping conditions in a foul atmosphere and almost incredible sweating were the inevitable result, and these conditions persisted until better heating and lagging arrangements could be introduced, particularly into smaller ships.

The fitting of urgent new equipment such as alternators for radar and D/G generators for degaussing involved siting these items often in positions on mess decks and flats which were quite unsuitable from the viewpoint of habitability. Even in arctic conditions, the amount of hot air discharged into the ship by the air cooling arrangements of these machines made living conditions most uncomfortable. It is considered most desirable that equipment of this type to be fitted in ships should be re-designed, perhaps with water cooling arrangements, and so make quite an important contribution to habitability below decks.

The full implications of long periods of "darken ship" in modern war conditions had certainly not been fully anticipated in peace, and ventilating arrangements, particularly for certain compartments such as galleys, were quite inadequate. In H.M.S. *Nelson*, for example, exhaust to the crew's galley was mainly by means of the open skylight, and when this skylight was closed for "darken ship" in the evening, conditions quickly became almost intolerable for the cooking staff and distinctly bad elsewhere in the ship. This was certainly a case where realistic exercises in peace would have anticipated most of these particular troubles in war.

The continuous running of steam machinery, particularly in the tropics, made conditions in machinery spaces particularly severe, and both temperatures and humidity rose above acceptable limits. The principal factors operating against tolerable conditions were, apart from ambient conditions, insufficient lagging of hot parts, inadequate and badly arranged ventilation, and steam leaks which raised the humidity.

As the tempo of the Japanese war increased it was realized that improvements in habitability conditions in ships destined for service in that theatre of war—and more particularly in East Indies waters—were essential. In 1944 a "Habitability Mission" made extensive on-the-spot investigations into the at-

tendant problems in the East Indies Fleet, and to some extent in the then recently formed British Pacific Fleet. The end of the war decreased urgency, but a valuable report was produced and, in conjunction with the investigations of the R.N. Personnel Research Committee, schemes for improvement in conditions were decided on and began to be put into effect.

Work is, of course, still going on in this direction, and much remains to be done, as it is of the highest importance. The whole story would provide sufficient material for—and deserves—a paper to itself; in fact, some of it has been published already. Improvements have been, and are being made in the following directions, with considerable betterment of habitability:—

(*a*) Improvement in living spaces and action compartments by more efficient ventilation, better fans, more air, better leads of trunking, careful siting of external intakes and exhausts, and consideration of the factors introduced by wartime "darken ship" and closed up conditions and air conditioning in selected spaces.

(*b*) Insulation of compartment boundaries, ventilation trunking and heat producing items to minimize the effects of wild heat.

(*c*) Improvements in layouts, in particular with regard to the location of heat-producing equipment to facilitate the removal of heated air produced by such equipment.

(*d*) In the machinery spaces introduction of more efficient steam pipe joints, and fitting of gland evacuation systems to reduce blow from turbine glands.

*Provision for "Amenities".*

The requirements of habitability under tropical conditions also forced a large increase in the provision of what may for convenience be called "amenities" but which are, in fact, essential for health and efficiency under such conditions, e.g. refrigeration for food storage, laundries and washing facilities, drinking water coolers, open air sleeping births, etc.

For ships keeping at sea for long periods in the tropics it is clearly necessary to have such things as adequate refrigerated storage in messes to keep such items as butter in good condition, e.g. the domestic type of refrigerator, and laundries provided with washing, drying, and ironing machines, not previously fitted in H.M. ships, for ensuring that personal cleanliness which is so essential in hot weather. It is also clear that adequate supplies of fresh water for washing and of cooled fresh water for drinking should be available.

The main problems were those of supply, fitting, and, when the machines were fitted, those of maintenance, and of giving personnel training for handling the plant. Nevertheless, before the war ended, a great deal had been accomplished in this direction and the work is still going on.

So far as the provision of sufficient fresh water to meet the maximum tropical requirements of ships' companies was concerned, it was found difficult to meet this in a considerable number of cases with the distilling capacity fitted, and the general lack of shore supplies of water. It was impracticable, both from supply and space considerations, to fit additional plant as an immediate remedy, and efforts were directed mainly to improving the availability and efficiency of existing plant and facilitating maintenance by provision of additional spare gear.

Most effort has been directed to methods of scale inhibition, and towards the end of the war some success had been achieved with starch injection.

Later investigations into this problem are of considerable interest, and again really deserve a paper to themselves, Apart from this, in new construction considerably increased distilling capacity is being fitted.

Besides the above, the appearance and

comfort of the living quarters of warships has seen many recent changes associated with improvements in furniture, lighting and decorative color schemes; the provision of tea bars, sandwich counters and improvements in the organization of food distribution have served to relieve much of the monotony formerly associated with feeding under crowded conditions, and in addition the more widespread use of sound reproduction equipment, additional educational facilities and more cinemas have all helped to improve life on board. Since the war, particularly, a lot has been done to improve bathrooms and washing facilities.

Lastly, towards the end of the Japanese war there was the birth of an ambitious project, the amenity ships, whose sole function was to provide, on a large scale, rest and recreational facilities which could not possible be provided in the ships on the station.

## Operational Activity In High Latitudes.

Operational activity in high latitudes brought problems, many of which had not been fully allowed for previously. Probably the most pressing concerned habitability below decks over long periods and the operational efficiency of weather deck personnel and weather deck equipment. In retrospect, it is not difficult to imagine the discomfort of being closed up for hours on end inside an armored turret in arctic conditions, but the end of the war still found some problems of this nature had not reached a fully satisfactory and practicable solution.

Problems of icing up of ropes, boats' falls, deck machinery and weather decks, particularly flight decks of carriers, were many and serious, and considerable effort and research were devoted to these questions. De-icing pastes were developed (some were already in commercial use) which certainly helped, but no really satisfactory solution for flight decks has yet been found, so far as is known.

One problem which could be troublesome and highly inconvenient in the midst of operational activity was the frailty of destroyers' forward shell plating against damage by ice. As is well known, a noticeable feature of Icelandic and Norwegian vessels is the steel sheathing fitted forward over stout wood planking and framing; the framing sometimes looks ridiculously heavy to British eyes, but centuries of experience have evidently shown its necessity, even for the slow speed of such ships. The danger to destroyers proceeding at wartime speeds in these waters was obvious but hard to cure. One expedient extensively tried was wood sheathing fitted internally in the region of the water-line, but this gave little protection to the plating and made temporary repair much more difficult in the event of splits in the shell plating and consequent leakage.

Icing up of exposed equipment or armament could, of course, be of vital consequence in war operations. H.M.S. *Jamaica,* in action off the North Cape damaged the muzzle ends of some of her forward gun-barrels though developing a slight internal coating of ice in the short interval the barrels were exposed before firing. The damage was sufficient to require renewal of the barrels, and corresponding action had to be taken in one destroyer whose guns had been similarly damaged.

Mere accumulation of ice on decks, superstructure, rigging, etc., could be a serious inconvenience in all ships, but in small ships, can, of course, be particularly dangerous. One trawler was lost in this way and special instructions had to be issued by the Admiralty for the guidance of officers in small vessels.

## MAINTENANCE.

Maintenance of ships and machinery in an operational condition throughout the years of a long war is always a pressing technical difficulty. Quite apart from the fact that shipbuilding and repairing resources are bound to be strained to the utmost with the work of new construction and repair of damage due to enemy action it is obviously of the utmost importance to a commander-in-chief that the maximum percentage of his force is always available for instant operation. For both these reasons it is imperative that a fleet's own resources with the help of any base repair staff available should be adequate to keep ships in the fighting line.

The authors would like to pay tribute here to the very great help rendered to the Home Fleet during the war by the small repair staff ashore at Lyness. The efforts of these men in all kinds of uncomfortable quarters afloat or ashore made a big contribution to the efficiency of the Home Fleet.

The volume of this maintenance effort was much increased by the great age of many of our ships. More lightly built than a mercantile vessel of corresponding size, many warships had to start out on a long and arduous war at an age when a shipbreaker's yard would have been the most logical next port of call. H.M.S. *Kent* saw as much of the stormy arctic seas as most ships and finished the war with some hundreds of bolts in place of rivets in the shell plating aft, all of which had of necessity to be fitted by divers with the ship at operational notice, since the ship could not reasonably be spared for a refit and docking further south. Although worse than the average, this sort of job was reasonably typical of the kind of work involved. It is perhaps reasonable to anticipate that all-welded hulls in future construction will suffer less from the weaknesses of old age than the riveted hulls of the past.

During inter-war years, when new ships were at a premium, the urge to press new wine into old bottles had many serious consequences. It resulted in increased draught, reduced stability, with increased vulnerability to underwater damage, increased wetness in bad weather with reduced ventilation at sea and loss of habitability, seriously overcrowded mess decks and washplaces, with increased strain on cooking facilities and difficulty in maintaining satisfactory arrangements for meals. Such ships are seldom fit to be exposed to front line risks in modern war and are generally bad investments.

Apart from the troubles due to age in older ships the hasty finish of warship new construction made necessary by the urgency of the war gave a lot of trouble to fleet and base repair resources. Water services, ventilation systems, and other equipment would expose deficiencies or develop defects after a short time on service. Some of the new *King George V* class battleships had hot water systems which were a genuine fifty-fifty mixture of copper and galvanized steel piping due to a complete bottling up of the supply of copper at a stage when it was impossible to cause serious delay in completion by replacing by steel all the copper already fitted. Ships would commission without water tube valves fitted at the weather deck to be closed at sea in bad weather; the effect of this omission on habitability at sea in an arctic convoy operation has to be experienced to be thoroughly appreciated.

Water services generally caused trouble in maintaining high efficiency due to corrosion of steel piping and impingement attack in the case of copper piping due to the proximity of gun-metal valves. Copper-nickel-iron piping is now being tried for fire mains and pure copper piping for fresh water services. This should reduce maintenance to a minimum and also, it is hoped, will tend to prevent internal fouling due to marine growths —a factor which has in some conditions a most serious effect on the efficiency of the fire main.

## Conclusion.

Experience in the last war and, even more, the probable conditions of any future war, make it imperative that hulls, machinery, and equipment must be designed with a view to the minimum need for and the maximum ease of maintenance, so that even the resources of any fleet train should be called upon as little as possible.

Also, at the risk of some repetition, it is desired to emphasize once more that years of peace inevitably result in some growth of weaknesses and operational inefficiency; this effect is accentuated by the overall need for national economy. Ruthless, thorough testing of our ships at sea—or at least one ship of each new class—within all reasonable limits of seamanship, and unaffected by considerations of economy, is a real necessity in the preparation of the Navy for any future war .

Finally, it should be appreciated that these notes, concerned as they are with troubles over many years of war, necessarily throw undue limelight on faults. A much more true overall picture lies in the fact that our ships gave magnificent service over many gruelling years and carried us successfully through the biggest war in history.

This paper is published with the approval of the Lords Commissioners of the Admiralty, but the responsibility for any statement of fact or opinions expressed rests solely with the authors.

# THE STRUCTURE OF NAVAL APPROPRIATION ACTS.

ACKNOWLEDGMENT.

The following timely article by Rear Admiral J. A. FURER, U.S.N. (Ret.) is reprinted from the "U. S. Naval Institute Proceedings" for December, 1948. Admiral Furer is a graduate of the Naval Academy in 1901, Rear Admiral Furer was in the Construction Corps until it was amalgamated with the Line. A pioneer in establishing scientific management in our navy yards, he has had many other interesting assignments, such as raising the submarine *F-4* from a depth of 304 feet off Honolulu in 1915. From November, 1941, until his retirement he was Coordinator of Research and Development for the Navy Department.

## *Introduction.*

The ever mounting cost of government is becoming a matter of great concern to all taxpayers. Not that there is anything new in the shaking of heads over the growth of such costs. Since the beginning of the Federal government an almost continuous study of this subject has been under way by committees of Congress, by special commissions appointed by various Presidents, and by private organizations. One of the declared objectives of the Hoover Commission appointed by President Truman in 1947 on the "Organization of the Executive Branch of the Government" is to find ways of "limiting expenditures to the lowest amount consistent with the efficient performance of essential services, activities, and functions" of the government. Inasmuch as the major items of the 40 odd billion dollar annual budget, such as the interest on the public debt, aid to veterans, foreign commitments, and certain other items, are not susceptible to appreciable contraction, it is not to be expected that the Commission will be able to find ways of making any considerable reduction in expenditures. Nevertheless any reduction that can be made is important because it is likely to lead to improving the operations involved.

A fundamental difficulty in reducing such expenditures is the lack in government of the profit motive as an incentive to efficiency and economy. In private enterprise management efficiency and management rewards from the top down are measured largely in terms of the ability to operate at low costs in order to make profits. In that field the determination of costs has therefore become of the greatest importance and accounting has become one of the most valued tools of management. The ultimate end of cost accounting in private business is to keep costs and selling prices in line. With this use of cost accounting the government is not normally concerned. The government should however be as much interested as private enterprise in the use of accounting as a tool of management for comparing operating efficiencies; for laying the foundation of sound budgetary panning; for exercising fiscal control at all levels of spending; and for measuring individual as well as group performance.

*Early Navy Cost Accounting.*

The Navy was a pioneer among the departments of the Federal government in recognizing cost accounting (as distinguished from appropriation accounting) as an essential instrument for measuring performance. Constructive interest in the subject began in the Navy when a movement under the name of "Scientific Management" got under way in private industry shortly after the turn of the century.

This philosophy of management was spearheaded by Frederic W. Taylor. The movement received much publicity and added impetus when Louis Brandeis, later a justice of the United States Supreme Court, expressed the opinion before a Congressional committee that scientific management could save the railroads of the United States a million dollars a day in operating costs. The efficiency experts, as Taylor's disciples were called, stressed the point that a large percentage of the failures in industry are due to insufficient knowledge of the costs of production. Cost determination was therefore placed very high on the list of the indispensables to good management. The principles of scientific management were taken up with great enthusiasm by certain elements in the Navy, particularly by the Construction Corps and by those members of the Supply Corps who were experts in accounting. Their interest lay largely in the money spent at navy yards and other industrial shore activities. Such expenditures represented then, as they do now, about one-half of the annual expenditures for the Navy. These men took the position that even though the government is not in business for profit, those responsible for the management of navy yards should know the cost of the work done at the yards in order to do it better and more economically.

When it was decided in 1909 to apply the principles of scientific management to the operation of navy yards, the setting up of a cost keeping system along modern industrial lines was included as an item of major importance. A firm of accountants was employed to lay out the system. These experts went about the job in the same way that they would have done for a large private industrial concern, only to find that naval appropriation acts, with their many subdvisions and the requirements of the government for strict appropriation accounting, made this impossible. It was found that appropriation accounting and cost determination had to be made independent processes, and that they could not be made identical without impairing the legality of the one or the correctness and administrative value of the other. A cost accounting system had therefore to be devised which is in a measure a makeshift, and which does not lend itself readily to the broader uses of accounting as a tool of management. Incidentally it may be mentioned that a clause was introduced in the Naval Appropriation Act for 1913, which has been repeated every year since then, making it illegal to pay the salaries of officers and civilian employees while engaged in making time or motion studies. This has handicapped the Navy Department greatly in applying some of the most worthwhile principles of scientific management to the operation of navy yards —such as the setting of standards and goals of performance. That, however, is another story and is not fundamentally involved in the structure of Naval Appropriation Acts.

The crux of cost determination, whether concerned with the production of physical things or of services, is the indirect expense involved in the processes of production. The costs of direct labor and of direct material are usually not difficult to determine, but these elements may be only a fraction of the total cost. The problem lies in ascertaining accurately and in distributing equitably the indirect, or so-called overhead expense. This difficulty increases with the size and complexity of the producing organization. It is great enough in private industry, but it is multiplied many times in Naval activities by the form in which Naval Appropriations are made. It will be helpful to an understanding of this difficulty to describe the appropriation system.

*Naval Appropriation System.*

In connection with making plans for carrying on their activities for succeeding fiscal years, the bureaus and offices of the Navy Department make estimates of the expenditures in money that will be involved. These plans and estimates are based on many considerations and on data obtained from many sources, including recommendations from the shore establishments and from the forces afloat. Each bureau has a fiscal division having as one of its functions the preparation of these estimates segregated under headings called "appropriations," the names of which more or less closely indicate the objects of the proposed expenditures. Explanations going into very great detail to justify each item of the proposed expenditures are prepared by the bureaus. Thousands of items go to make up the final estimates.

From the bureaus and other subdivisions of the Navy Department the estimates go to the Director of Budget and Reports of the Navy Department, where they are further reviewed and coordinated. The authority and responsibilities of that office and of the Fiscal Director of the Navy Department have been greatly enlarged in the last few years. It is the function of the Director of Budget and Reports to formulate, supervise, and coordinate all policies and procedures affecting the budgetary activities of the Navy. This includes prescribing the form and method of preparing appropriation estimates, reviewing program planning to insure adequacy for budget formulation, and the actual preparation and presentation of the Navy's budget. Representatives of the office make field inspections, hold hearings within the Naval service, and review and evaluate the estimated financial requirements in terms of the missions to be performed by the respective components of the Naval establishment. The Fiscal Director formulates and coordinates the procedures affecting the financial activities of the Navy, and prescribes the type and content of all accounting and financial records. Broadly speaking, the Fiscal Director keeps track of Navy money *after* it is appropriated.

Until 1921 the appropriation estimates, prepared in principle but not so elaborately as described above, went from the Secretary of the Navy to the Secretary of the Treasury for transmission to Congress. Since the establishment of the Bureau of the Budget in 1921, all appropriation estimates are submitted to that bureau for review before going to Congress. One of the purposes in establishing that office was to provide a disinterested agency for reviewing and revising the appropriation estimates of the executive departments. This has in fact become one of its principal activities, rather than just the preparation of an annual budget according to the usual definition of that term. Justification of estimates before the Bureau of the Budget has become a routine year round activity of the fiscal divisions of the bureaus and of the office of the Director of Budget and Reports.

The estimates for the Navy, together with those of all other federal activities, are assembled in an annual report by the Bureau of the Budget for transmission to Congress by the President. Various committees of Congress then call on representatives of these activities to appear before them to explain the purposes for which the money is to be used and to justify the amounts requested. In the case of the Navy estimates, the most searching hearings are those held by the subcommittee on appropriations for the Navy Department of the Committee on Appropriations of the House of Representatives. The reason is that the Naval Appropriation Act, as presented to the House of Representatives, originates in that subcommittee. The scope of these hearings may be judged from the fact that in connection with the Naval Appropriation Act for 1948, some 200 naval officers and civilians from

the Secretary of the Navy on down appeared before that subcommittee alone, and that these hearings take up a volume of 1819 printed pages totalling almost a million words.

Congress is not bound to follow the appropriation estimates recommended by the Budget Bureau, and it has not yielded its prerogative of having the final word on how much money is to be appropriated. Increases in the estimates originally submitted are not infrequent. The Naval appropriation act as finally passed by Congress is based not only on a great number of technical and administrative considerations on which millions of words of explanation and justification have been lavished, but also on the political considerations of the moment revolving around broad questions of national policy.

*Obligating Authority.*

It is often asked, "Just what is an appropriation?" An appropriation by Congress is a *law,* having the full force and effect of any other law. It defines the purposes for which the money appropriated can be used. Further, it stipulates the amount of money which can be *obligated* for these purposes. The term "obligated" means the right to commit the Secretary of the Treasury to the payment of money out of the treasury for the purposes authorized by the appropriation act. The Navy Department actually has five kinds of obligating authority stemming from appropriation acts:

(1) Obligating authority contained under the so-called annual appropriation titles. The principal appropriation titles in this category are "Aviation, Navy"; "Maintenance, Bureau of Ships"; "Ordnance and Ordnance Stores"; and so forth for the various bureaus, Marine Corps, and certain other sub-divisions of the Navy Department. Obligations may be entered into under this authority after the bill is signed until June 30 of the fiscal year covered by the act. The Navy has 24 months from the end of the fiscal year to clear bills and to make payments for the obligations incurred.

(2) Contract authority is sometimes provided to supplement an annual appropriation. So far as obligating the Treasury is concerned, contract authority is identical with the annual appropriation under the same title. If an appropriation and additional contract authority are included in the act under the same appropriation title, the sum of the two figures is the amount that the Navy Department may obligate during the fiscal year in question. The distinction is that contract authority permits the *obligation* of funds, the cash to pay which will be appropriated subsequently.

(3) Obligating authority under an appropriation title is sometimes granted by transferring a balance from some other appropriation title or fund. The transfer is usually made from an item or a fund that is in excess of requirements, to a title that is short of estimated requirements.

(4) Continuing appropriations are those identified in the Appropriation Act by the words "to remain available until expended," and are employed under such titles as "Construction of Ships, Navy," "Ordnance for New Construction," and "Public Works." There is no time limit on clearing bills and making payments under these titles as there is in the case of the annual appropriations. Such appropriations also usually provide the authority to make obligations for the *entire cost* of programs, although the expenditures to complete those programs will continue over several years.

(5) The Navy has four principal revolving funds which carry obligating authority: the naval stock fund and the clothing and small stores fund that have been set up with money specifically so earmarked in appropriation acts; the working fund fed by deposits from other government departments and private parties which was created in order to simplify doing work for such customers and for obtaining reimbursement for the expenditures made; and the procurement fund fed by deposits from the annual appropriations. The obligating authority which the Navy has under the four funds is the total value of the funds, but as of any given moment it is the unobligated balance of the funds.

*The Bureau System.*

The structure of Naval Appropriation Acts goes back in principle to the reorganization of the Navy Department on the Bureau system in 1842, but changes in the degree of detail covered by the acts have been almost continuous since that time. The principle underlying the bureau system is the segregation and distribution of the Navy Department's technical and administrative work along functional lines. Under the law establishing this organization, the Secretary of the Navy was authorized to distribute the work of the Navy Department to the bureaus. To a certain extent he had a free hand in allocating to the bureaus the funds appropriated annually by Congress for the Navy. The system did not remain static very long. In the hundred odd years since its adoption, the growing complexities of the Navy, resulting from the scientific and technological advances of the machine age, have continually brought into being new purposes for which money has had to be provided. Additional bureaus have been set up to handle some of these developments; the scope of the activities of all of the bureaus has been vastly enlarged; the functions of the bureaus have been reshuffled many times; and there have been split-ups, mergers, and even one case of the complete abolition of a bureau, with transfer of its functions to the remaining bureaus. In fact the Bureau of Medicine and Surgery is the only bureau that has remained the same in name and in function since it was established in 1842.

With the inauguration of this system, the Bureau of Navy Yards and Docks was charged with the construction and maintenance of the docks, wharves, and buildings at the navy yards, and was given general control of the administration of the yards. The commandants of the navy yards reported to the chief of that bureau. Since then the Navy has gone through a complete cycle in the matter of bureau control and responsibility for the management of its shore establishments, particularly of navy yards. The control of the Bureau of Navy Yards and Docks (later called the Bureau of Yards and Docks) over the navy yards gradually declined, and the influence of the other bureaus in the affairs of the yards gradually increased until all reached about the same level of authority in this respect.

By the end of the century the point was finally reached where each of the five technical bureaus operated a complete set of shops and had a power plant of its own at each of the principal navy yards. This duplication of facilities and of administrative effort became the subject of much discussion and criticism in Congress and in the Navy itself—especially by the advocates of scientific management already mentioned — with the result that consolidation of the shops and of management were brought about beginning with what was called the Newberry reorganization in 1909. During the period that followed the bureaus were in theory completely divorced from any authority or responsibility for the management of the navy yards. A division was set up in the Secretary's office to handle such matters. The bureaus retained technical responsibility for the work of the shore establishments, decided what work was to be done, and provided the money by allotments from their appropriations for doing the work. They also had the principal responsibility for staffing such establishments with competent specialist officer personnel. Actually the bureaus continued to exercise considerable management control over the navy yards through such personnel. This policy stood the test of World War I and of the following peace period, but it had obvious shortcomings. Toward the end of World War II the bureaus were again assigned management sponsorship and responsibility for the various shore establishments, but the responsibility was placed on a single bureau for each activity. The Bureau of Ships, for example, has management sponsorship for naval ship yards, the Bureau of Ordnance has management sponsorship for the Naval Gun Factory and for ammunition depots, the Bureau

of Personnel has management sponsorship for naval training stations etc.

Whatever other effect on the administration of the Navy the fluctuation of bureau influence in the affairs of the navy yards may have had, it resulted definitely in proliferation of the naval appropriation structure. In their desire to define their functions more positively and to tighten their grip on matters in which they were interested the bureaus often requested Congress to make specific appropriations to cover such matters. Congress was usually not loath to make appropriations under specific titles because the practice appeared to strengthen the hold of the committees on the purse strings. Therefore the pattern of appropriation acts became a main or leading appropriation covering each bureau's activities, with numerous additional titles more specific in their nature. In most cases the objects of expenditure mentioned in the specific appropriations were covered also in general terms by the main appropriation, so that often payment could be made from the bureau's main appropriation as well as from the specific appropriations. In addition there were titles covering in detail the maximum number of white collar workers (clerks, draftsmen, and other technical personnel) that could be employed in each of the shore establishments, bureaus, and subdivisions of the Navy Department, and the maximum pay that each individual could receive.

The bureaus adopted the practice of controlling their appropriations by budgeting them at the beginning of the fiscal year, retaining a sizeable reserve for unforseeable contingencies. Allotments on a monthly or quarterly basis were made to the various spending activities, or to cover specific projects. Periodic returns of actual expenditures from these activities provided information as to balances available. If, as the fiscal year advanced, a bureau found itself with a growing margin of funds over probable needs, the allotments were increased or additional projects were undertaken that had been given lower priority at the beginning of the year. These are still the principal steps in keeping track of appropriations, but spending discretion and authority have been decentralized to a considerable extent. For example, decisions as to what repairs on ships to undertake at naval shipyards is now left to the forces afloat within the limits of alloted funds, thus providing an incentive for more self-maintenance and for greater discrimination in the work requested of the naval shipyard. A ceiling has been set on the total numbers of personnel that may be employed at the respective industrial shore establishments, even though funds may be available to employ temporarily a larger force. The number of white collar workers, supervisors, and other non-producers is regulated by specific allotment of funds for that purpose only. However, no matter how carefully a bureau manages its funds under the present appropriation structure, there is no assurance that all work can be undertaken in the order of its importance as seen by the bureaus. For example, the Bureau of Ordnance and the Bureau of Ships may have set aside the money for certain very important underwater explosion experiments involving the use of tugs and the presence of bureau technical personnel. However, when the time comes for making the experiments, it may not be possible to undertake the work because there is no money available for manning the tugs and to cover the travelling expenses of the technical personnel, as the funds for these purposes come under the cognizance of another bureau.

*Proposed Revision.*

The number of appropriation titles increased gradually over the years until there were upward of 200 separate items in the annual naval appropriation acts just after World War I. The great expansion in Navy spending with the outbreak of World War II, and a shortage of accounting personnel, soon caused

appropriation accounting to fall so far behind that it was of little value in furnishing up-to-date information as to the status of the appropriations. In order to provide relief from this situation most of the specific appropriations were eliminated, and lump sum limits were inserted in the main appropriations of the bureaus to cover the pay of white collar workers. These steps, and the introduction of more mechanical bookkeeping appliances, have put appropriation accounting back on its feet, but cost accounting for purposes of management and fiscal control stands about where it was before. A simplified appropriation structure to correct this trouble has been recommended many times in the past by the Bureau of Supplies and Accounts and by others. The subject is again before Congress on the recommendation of the Secretary of the Navy. The handicap to satisfactory cost accounting imposed by the present acts lies in the fact that much of the indirect expense involved in doing any item of work at naval shore establishments must be charged to a considerable number of appropriations. Once so made, the charges cannot conveniently be recaptured for inclusion by simple accounting procedures in the cost of the work. This in turn impairs the usefulness of accounting as a tool of management.

The accounting difficulty, however, is only one of the aspects of the unsatisfactory management position in which the Navy Department is placed by reason of the numerous appropriations to which every primary Navy function is charged. The situation may be summarized as follows. The present appropriation structure involves all of the bureaus in the fiscal affairs of numerous activities over which they have very little fiscal control, inasmuch as practically every expenditure that originates with any one of the leading appropriations makes some call, direct or indirect, on the other appropriations. This is true even when the major part of the cost is absorbed by the appropriation having primary cognizance over the work. The contributing appropriations thus cut across functional and organizational lines, and there is always overlapping of the appropriations at the functional boundaries. This results in diffused fiscal responsibility, in awkward and complicated internal management, and in difficulty in making effective budget presentations. Fiscal responsibility is diffused because undertakings are financed from many sources and therefore have many fiscal masters. Fiscal control is consequently divorced from management responsibility. Management is complicated because the bureaus are involved in the fiscal affairs of numerous activities over which they do not have primary management control. Effective budget presentation is difficult because the money requirements for contemplated undertakings are presented piecemeal, and because the estimates under the contributing appropriations are justified by bureaus which do not have definite management responsibility for the expenditures. These handicaps make it difficult to present clear and concise estimates to the Bureau of the Budget and to Congress.

The Fiscal Director of the Navy Department recently prepared a number of diagrams to show how the present Naval appropriation system works. The Bureau of Medicine and Surgery and a naval hospital rather than the more intricate example of a naval shipyard were chosen to illustrate the working of the system. Fig. 1 shows the flow of funds and the returning reports. Fig. 2 shows the Bureau of Medicine and Surgery involved in the fiscal affairs of all of the shore establishments, but with *primary* management control over hospitals only. Fig. 3 shows that all bureaus are involved in the fiscal affairs of all of the

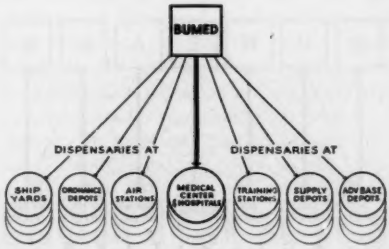



FIG. 2. Under the Present Act.

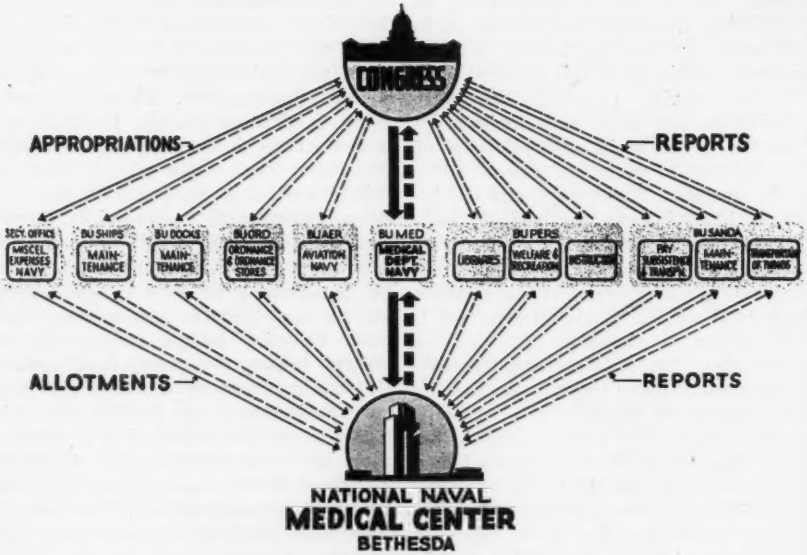



FIG. 1. Under the Present Act.

shore establishments. A diagram similar to Fig. 3, showing appropriation titles and objects of expenditures, could be prepared showing how all appropriations contribute something to each expenditure object. The crisscross of lines would make the difficulty of accurate cost determination even more striking.

Efficiency and economy would be promoted if the present appropriation structure were revised by organizing appropriations in terms of primary functions, by eliminating contributing appropriations (except for naval personnel, inasmuch as separate financing of such personnel is not illogical), and by describing purposes under the appropriation titles in such language as to avoid overlapping. Accounting procedures could then be set up to more accurately satisfy both appropriation accounting and cost accounting. One of the steps to bring about this reform is the recasting of the appropriation titles. As a beginning in this direction, the Navy Department has proposed the following for the 1950 Appropriation Act:

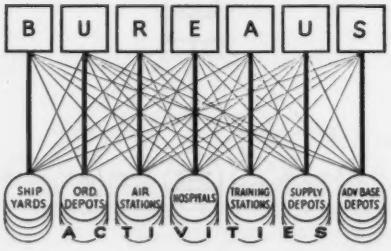



FIG. 3. Under the Present Act.

*Office of the Secretary.*
- Miscellaneous Expenses
- Claims, Navy
- Contingencies of the Navy
- Operation and Conservation of Naval Petroleum Reserves
- Salaries, Office of the Secretary of the Navy
- Salaries, Office of Judge Advocate General
- Contingent Expenses
- Printing and Binding

*Bureau of Ships.*
- Maintenance, Bureau of Ships
- Construction of Ships
- Increase & Replacement of Naval Vessels—Construction & Machinery
- Salaries, Bureau of Ships

*Office of Chief of Naval Operations.*
- Island Governments
- Naval Observatory
- Hydrographic Office
- Salaries, Office of Naval Records and Library
- Salaries, Office of Chief of Naval Operations
- Salaries, Board of Inspection and Survey
- Salaries, Office of Chief of Naval Communications
- General expenses

*Bureau of Ordnance.*
- Ordnance & Ordnance Stores, Navy
- Ordnance for New Construction
- Increase & Replacement of Naval Vessels—Armor, Armament & Ammunition
- Salaries, Bureau of Ordnance

*Office of Naval Research*
- Research, Navy
- Salaries, Office of Naval Research

*Bureau of Supplies and Accounts.*
- Subsistence, Navy
- Maintenance, Bureau of Supplies and Accounts
- Transportation of Things
- Fuel, Navy
- Salaries, Bureau of Supplies and Accounts

*Bureau of Medicine and Surgery.*
- Medical Department
- Salaries, Bureau of Medicine and Surgery

*Bureau of Aeronautics.*
- Aviation, Navy
- Construction of Aircraft & Related Procurement, Navy
- Salaries, Bureau of Aeronautics

*Bureau of Naval Personnel.*
- Training and Education, Navy
- Welfare of Naval Personnel
- Officer Candidate Training
- General Expenses, Bureau of Naval Personnel
- Naval Reserve
- Naval Academy
- Naval Home, Philadelphia, Pa.
- Pay and Allowances of Naval Personnel
- Retired Pay, Navy
- Transportation and Recruiting of Naval Personnel
- Salaries, Bureau of Naval Personnel

*Bureau of Yards and Docks.*
- Maintenance, Bureau of Yards and Docks
- Public Works, Bureau of Yards and Docks
- Salaries, Bureau of Yards and Docks

*Marine Corps.*
- Pay, Marine Corps
- Retired Pay, Marine Corps
- Pay of Civil Force, Marine Corps
- General Expenses, Marine Corps

As a necessary part of the proposed revision in appropriation structure, some redistribution of bureau cognizance will be involved. Funds to cover service and maintenance functions would be transferred to the bureau having management responsibility for the activity. This would result in fiscal responsibility going hand in hand with management responsibility. Each activity would thus have a single fiscal master to be held accountable for results by the Secretary of the Navy and by Congress. Fig. 1 would then become Fig. 4, and Fig. 3 would become Fig. 5. Management of the respective establishments would be simplified both for the bureaus and locally.

A simple illustration will clarify the proposed revision. The naval training station at Great Lakes, Illinois, is under the management sponsorship of the Bureau of Personnel. Under the present appropriation system, the small boats used at the station for training and transportation purposes are supplied and repaired under appropriation "Maintenance, Bureau of Ships." Under the new arrangement the Bureau of Ships would still have the responsibility for the design and procurement of such craft (either by manufacture at naval shipyards or by purchase), but the Bureau of Personnel would pay for the

FIG. 4. Under the Revised Act.

boats and their unkeep from its appropriation "Training and Education," and the charges would appear as items of the station's operating costs for camparison with similar costs at other training stations. Estimates and justifications for the proposed expenditures would appear in the budget of the Bureau of Personnel.

The control by Congress over the size of appropriations and over the purposes for which they may be spent would not be weakened by reducing the number of titles in Naval Appropriation Acts. In fact, control would be strengthened through simplification and crystallization of budget presentations. Each bureau would have to prepare and defend the complete estimates for the activities under its cognizance, and it could therefore be held solely responsible for the money it receives and spends. In the realm of accounting, Congress is primarily interested in appropriation accounting and in auditing. These would remain as important and indispensable as ever; but the sights would be raised, and the terms of accountability would extend to accounting for operating results and to the measurement of performance against goals or standards initially set through budgetary planning and cost estimates.

In some quarters objections have been made to the proposed changes on the grounds that placing management sponsorship and fiscal control in the hands of a single bureau for each activity will sacrifice some of the advantages of the Bureau system. For example, the Bureau of Yards and Docks and the Corps of Civil Engineers are specialists in the maintenance of shore facilities and in the operation of power plants. It is asked, "will the organization of a naval shipyard function satisfactorily if the division handling such work is manned

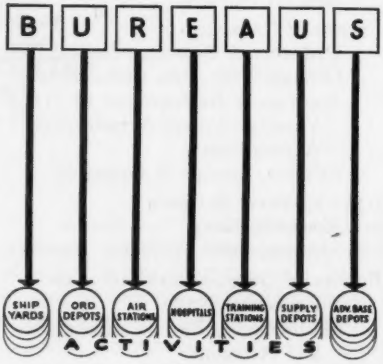



FIG. 5. Under the Revised Act.

by officer personnel owing primary allegiance to the Bureau of Ships while on such duty, but whose careers in the matter of promotion and assignment to future duty are largely in the hands of the Bureau of Yards and Docks?" The Bureau of Supplies and Accounts and the Supply Corps are specialists in storekeeping, accounting, and transportation, among other things. Will such duties be performed as satisfactorily at naval shipyards by officers working under similar conditions of divided self interest and allegiance? It is hardly to be supposed

that the Bureau of Ships would provide a group of specialists, separate from the Civil Engineer and Supply Corps, for managing the public works and supply division at naval ship yards. Basically such a policy would be unsound, because it would deprive the officers of these two corps of much of the training and experience they need to discharge the broader functions of their bureaus. Complete responsibility and cognizance of the Bureau of Yards and Docks over new public works, and of the Bureau of Supplies and Accounts over its duties other than certain service functions, would of course remain unchanged.

Fifty years ago the shibboleth of corps loyalty would have carried more weight than it does today. During these years the strong feeling of corps solidarity has abated considerably. The recognition of the soundness of the organizational device of having administrative lines of responsibility differ from functional lines in the operation of land, sea, and air forces in order particularly to achieve unity of command has had much to do with this change in feeling. Once this principle is also more completely accepted in the operation of the Navy's shore establishments, it will be recognized that the appropriation of funds should parallel the administrative rather than the functional lines of responsibility. The result will be that outstanding officer specialists from any branch of the Navy will have the opportunity of participating to a larger extent, and of reaching positions of greater responsibility, in the management of naval shore activities under the proposed appropriation structure than under the present structure.

# MODERN PRACTICE IN MARINE REFRIGERATION.

ACKNOWLEDGMENT.

This article by S. W. BROWN, Chief Engineer, Marine Department, Carrier Corp., is reprinted from "The Nautical Gazette" for January, 1949. It is a condensed version of an address delivered before the Institute of Food Technologists, Western New York Section, October 30, 1948, in Lyman Hall, Syracuse University.

The maladjustment of sections of the world's population to the available food supply is receiving increased attention. The world demand for refrigeration is greater now than ever before in history. India's old famine menace, the loss of seed between crops because of the devastating heat and humidity, is being solved by the installation of refrigerated warehouses in the cities of Lahore, Delhi and Gurdaspur, the first step of a long range program. A second step to increase India's food supply is a program for the quick freezing of fish. The initial installation, which is being built by the Government as a pilot plant to be turned over later to private enterprise is now being constructed in Bombay. Because of the high temperature and lack of refrigeration facilities in India, it is estimated that nearly one-half of the annual catch is wasted through spoilage. Another important factor which is contributing to the alleviation of food shortages is the refrigerated cargo vessel which makes possible the transportation of perishable foods from areas of abundance to areas of want.

The use of mechanical refrigeration for the preservation of food on shipboard was first introduced successfully in the 1880's on a ship of British Registry for the transportation of frozen beef. Today, practically every ocean-going vessel is equipped with a mechanical refrigeration plant. The application of refrigeration introduced to the steamship owner a new source of income through the transportation of refrigerated cargo. Considerable differences in practice exist, not only in each maritime nation, but also among companies engaged in the same trade. Until recently no attempt had been made in this country to standardize the application of mechanical refrigeration installation on shipyard. A proposed standard which has been in the making for the last three years is now awaiting final approval by the American Society of Refrigerating Engineers.

The following are among the reasons why marine refrigeration differs from shore practice:

1. The installation should be designed to function properly under conditions of roll and pitch, to which a ship is normally subjected.
2. The equipment should be suitable to withstand normal usage under the corrosive effects of sea air and sea water.
3. The installation should be designed to suit the physical arrangement of the ship.
4. The installation should be designed to operate through the complete range of climatic conditions through which the ship must travel and be capable of adjusting itself to sudden changes frequently encountered in the ambient sea water temperature.

5. Except in the case of a single-purpose vessel, such as a meat carrier or a banana ship, cargo refrigeration installations should be provided with refrigerating equipment capable of preserving cargo of every nature.
6. Since ships enroute cannot be easily serviced even from land based planes, it is essential that the system be designed for uninterrupted operation during the voyage. Accordingly, stand-by plants or easily replaceable spare parts should be furnished.
7. In view of the high premium for cargo space, the equipment should be designed to occupy a minimum of space, consistent with reliability and cost.
8. The equipment should be designed to conform with the applicable rules and regulations of various regulatory agencies, such as the U. S. Coast Guard, the U. S. Public Health Service, the American Bureau of Shipping, Lloyd's Registry, U. S. Maritime Commission, American Institute of Electrical Engineers, and the U. S. Navy.
9. In addition to the above, in the case of Naval vessels, many stringent requirements must be complied with, such as shockproofness, which further complicates the design of marine refrigeration plants.

Marine refrigeration has two general classifications: (1) Ships' Stores and (2) Cargo Refrigeration.

*Ships' Stores.*

The ships' stores plant provides the refrigeration for the compartments containing the perishable foodstuffs required for the feeding of the crew and passengers during the voyage. Occasionally this plant also provides the refrigeration for ice-makers, drinking water and galley-boxes.

Ships' stores compartments are subdivided to provide at least one high temperature and one low temperature. Table 1 shows the various compartments and temperatures provided on four classifications of U. S. Cargo Vessels and on Victory and Liberty Ships:

Valuable experience was gained during the war regarding the care of perishables in ships' stores. Serious spoilage of foodstuffs resulted in a directive being issued in the closing years of the war by the War Shipping Administration, requesting the reduction of box temperatures as follows:

Egg and cheese box (formerly dairy box)—30 to 31° F., never below 30° F.

Vegetable box—32 to 34° F.

Meat box — The temperature shall be set as low as possible (0° F. is recommended.) Under no circumstances shall the temperature be held above 10° F.

**TABLE 1: TEMPERATURE—DEGREES F**

| Space | C1 Vessel | C2 Vessel | C3 Vessel | C4 Vessel | Liberty | Victory |
|---|---|---|---|---|---|---|
| Meat ..... | 22 to 25 | 20 | 15 to 20 | 15 to 20 | 15 to 20 | 15 |
| Fish ..... | 20 | .. | 15 to 20 | .... | 15 to 20 | 15 |
| Fruit & Veg. | | | | | | |
| Veg. ..... | 35 to 40 | 40 | 42 to 45 | 40 to 45 | 40 to 45 | .. |
| Meat or Veg. .... | .... | .. | .... | 15 to 20 | .... | .. |
| Dairy ..... | 32 to 40 | .. | 35 to 40 | .... | 30 to 35 | 35 |
| Thaw ..... | 40 to 50 | 45 | 40 to 45 | .... | .... | 40 |

Fish box—The temperature shall be held as low as possible. (0° F. is recommended.) Under no circumstances shall the temperature be held over 10° F.

All post-war vessels now specify lower temperatures. The S.S. *Excalibur,* one of the "Four Aces" of the American Export Lines, is provided with ships' stores compartments capable of maintaining 0° F. Table 2 shows the ships' stores spaces and temperatures provided on this vessel:

**TABLE 2: SHIPS' STORES ON THE *S.S. EXCALIBUR***

| Space | Gross cu. ft. | Temperature |
|---|---|---|
| Meat Outbound | 1,654 | 0° F. |
| Meat Return | 1,090 | 0° F. |
| Poultry | 562 | 0° F. |
| Frozen Food | 540 | 0° F. |
| Fish | 577 | 0° F. |
| Ice Cream | 435 | 0° F. |
| Butter and Shortening | 772 | 15° F. |
| Ice | 175 | 20° F. |
| Eggs, Cheese and Milk | 1,290 | 35° F. |
| Potatoes | 1,070 | 35° F. |
| Fruit and Vegetables | 1,560 | 35° F. |
| Thaw | 2,640 | 45° F. |
| Total Cu. Ft. (Gross) | 12,365 | |

*Ships' Stores High Side.*

Ships' stores plants are usually designed for direct expansion, Freon-12. A few installations on passenger vessels have employed brine circulation when equipped with cargo refrigeration also using brine.

With very few exceptions, Freon-12, motor driven, reciprocating compressors are employed, with shell-and-tube sea water condensers. The size of the refrigerating plants vary from 3 to 5 H.P. for small vessels to over 100 H.P. for super-liners similar to those recently proposed for the U. S. Lines. For the smaller plants one compressor is furnished, with complete stand-by as a spare. On the larger plants, it is usually desirable to furnish two or three compressors to carry the load and one additional compressor as a complete standby. Where cargo refrigeration or air conditioning is included, it is sometimes possible to arrange the installation so that one compressor serves as a stand by for all three plants.

*Ships' Stores Low Side.*

Evaporators for ships' stores compartments fall into three categories:

1. Air coolers, usually of ceiling suspended type, for temperatures of 35° F. and above.
2. Prime surface coils for compartments maintained below 35° F.
3. Plate type evaporators.

It has not been until recently that plate type evaporators have been installed on shipboard. They are usually arranged so that the plates form shelving which makes it particularly applicable for quick frozen food packed in uniform cartons.

### *Cargo Refrigeration.*

Cargo Refrigeration is more complex because the ship owner requires an installation which provides sufficient flexibility to carry any type of perishable cargo and capable of maintaining any temperature from —10° F. to +55° F. Except for the completely refrigerated ship, most refrigerated cargo vessels are provided with multiple compartments accessible through doors leading to the hatch and confined to 'tween deck spaces. The compartments vary in number and size depending upon the requirements of the trade.

### *Cargo High Side.*

The high side equipment usually consists of multiple reciprocating Freon-12 compressors. In the case of some of the more modern installations, where the tonnage is sufficiently large, Freon-11 centrifugal refrigerating machines are used.

The value of a single cargo often exceeds the cost of the entire refrigeration plant. Accordingly, regardless of the number of compressors required to carry the load, at least one compressor is provided for complete stand-by.

Electric motors are predominately used with reciprocating compressors. Steam turbines have been applied to good advantage on centrifugal machines, which are particularly adapted to this method of drive, since its operating speed varies from 4000 to 8000 R.P.M. Until recently, Freon-12 reciprocating compressors were driven through multiple V-belt drive. Today compressors are designed to operate at 1750 R.P.M., making direct connection to motors possible and thus resulting in a substantial decrease in space and weight.

### *Cargo Low Side.*

In the case of cargo refrigeration, there are several types of low side equipment employed. Today the most widely used design for the all purpose cargo vessel is the diffuser system. This consists of a fan coil assembly, either of the factory self-contained type or the built-up type. Extended surface cooling coils are commonly employed because of the savings in space and the reduction of refrigerant charge. Defrosting is most effectively accomplished by sea water spray for direct expansion systems, and, in the case of indirect systems, it is the usual practice to circulate warm brine through the coils.

In the diffuser system the conditioned air is normally distributed through the sheet metal ducts, usually located around the periphery of the compartment and designed to blow either overhead, under gratings, or both, depending upon the nature of the cargo. The outlet slots for overhead distribution are placed near the ceiling. Bottom slots are also provided near the bulkhead to allow the air to blow down along the wall between the cargo battens and flow across the deck under the floor gratings. With this method, the cargo is completely blanketed with a constant circulation of conditioned air.

A second method is known as the wall coil type. This consists of zig-zag or hair-pin prime surface coils installed along the walls of the compartment enclosed in a solid baffle running from deckhead to the floor gratings. A sheet metal duct provided with bottom slots is run around the periphery of the walls above the cooling coils. A re-circulating air blower forces the air through the duct work, which is so arranged that the air discharges vertically downward over the prime surface coils, where it is cooled and flows along the deck underneath the high floor wood gratings. The conditioned air then rises through the cargo

at a relatively low velocity and is returned to the blower, which is usually suspended from the deckhead or ceiling. For direct expansion systems hot gas defrosting is normally employed. Where brine is used warm brine is circulated through the coils when defrosting is required.

With banana cargo, a third type of system is preferable. This is identical to the diffuser system except for the means of air distribution. A solid baffle or apron, as it is sometimes called, running from deck to deckhead, is constructed around the periphery of the compartment. Adjustable air outlets are provided both along the top and bottom of the apron, which can be adjusted to suit the type of cargo being carried. In this system it is possible to distribute all of the air through the bottom outlets for under grating circulation or through the top outlets for horizontal overhead distribution. When bananas are carried, it is the usual practice to distribute all of the air underneath the floor gratings, permitting it to filter up at a low velocity through the bananas and back to the fan room through a high return opening located near the deckhead.

*Recommended Temperatures.*

The homogeneous pre-cooled cargoes are usually carried at the same temperatures recommended for shoreside storage. When it is necessary to carry mixed cargoes in the same compartment, a temperature must be determined which will be suitable for all of the perishables. Table 3, which has been taken from the Applications Volume, Refrigerating Data Book (Second Edition) of the American Institute of Refrigerating Engineers, pages 124-5, shows the most favorable conditions recommended for warehouse storage and is used as a guide for marine transportation.

## TABLE 3: STORAGE REQUIREMENTS AND PROPERTIES OF PERISHABLE FOOD.

| Commodity | Storage Temperature, F | Relative Humidity, % | Approximate Storage Life | Water Content, % | Average Freezing Point, F |
|---|---|---|---|---|---|
| Apples | 30-32 | 85-88 | Fig. 1 | 84.1 | 28.4 |
| Apricots | 31-32 | 80-85 | 1- 2 wk. | 85.4 | 28.1 |
| Artichokes (Globe) | 31-32 | 90-95 | 1- 2 wk. | 83.7 | 29.1 |
| Artichokes (Jerusalem) | 31-32 | 90-95 | 2- 5 mo. | 75.5 | 27.5 |
| Asparagus | 32 | 85-90 | 3- 4 wk. | 93.0 | 29.8 |
| Avacados | 40-55 | 85-90 | — | — | 27.2 |
| Beans (Green or snap) | 32-40 | 85-90 | 2- 4 wk. | 88.9 | 27.7 |
| Beans (Lima) | 32-40 | 85-90 | 2- 4 wk. | 66.5 | 30.1 |
| Beets (Bunch) | 32 | 85-90 | 10-14 days | — | — |
| Beets (Topped) | 32 | 95-98 | 1- 3 mo. | 87.6 | 26.9 |
| Blackberries | 31-32 | 80-85 | 7-10 days | 85.3 | 28.9 |
| Broccoli, sprouting | 32-35 | 90-95 | 7-10 days | 89.9 | 29.2 |
| Brussels sprouts | 32-35 | 90-95 | 3- 4 wk. | 84.9 | — |
| Butter | −10- 0 | — | — | 16.0 | Approx. 28.0 |
| Cabbage | 32 | 90-95 | 3- 4 mo. | 92.4 | 31.2 |
| Carrots (Bunch) | 32 | 85-90 | 10-14 days | — | — |
| Carrots (Topped) | 32 | 95-98 | 4- 5 mo. | 88.2 | 29.6 |
| Cauliflower | 32 | 85-90 | 2- 3 wk. | 91.7 | 30.1 |
| Celeriac | 32 | 95-98 | 3- 4 mo. | 88.3 | — |
| Celery | 31-32 | 90-95 | 2- 4 mo. | 93.7 | 29.7 |
| Cheese | Approx. 34 | 65-70 | — | 37-38 | Approx. 28.0 |
| Cherries | 31-32 | 80-85 | 10-14 days | 83.0 | — |
| Chocolate candies | 68-70 | 50-55 | 6-10 mo. | — | — |
| Corn, sweet | 31-32 | 85-90 | — | 73.9 | 28.9 |

| Commodity | Storage Temperature, F | Relative Humidity, % | Approximate Storage Life | Water Content, % | Average Freezing Point, F |
|---|---|---|---|---|---|
| Cranberries | 36-40 | 85-90 | 1- 3 mo. | 87.4 | 27.3 |
| Cucumbers | 45-50 | 80-85 | 10-14 days | 96.1 | 30.5 |
| Currants | 32 | 80-85 | 10-14 days | 84.7 | — |
| Dewberries | 31-32 | 80-85 | 7-10 days | — | 29.2 |
| Eggplant | 45-50 | 85-90 | 7-10 days | 92.7 | 30.4 |
| Eggs, Shell | 29-31 | 85-90 | 9 mo. | 67.0 | Approx. 28.0 |
| Eggs, Shell, farm cooler | 40-55 | 75 | — | 67.0 | Approx. 28.0 |
| Eggs, Frozen, Freezing | −20 | — | — | 73.0 | Approx. 28.0 |
| Eggs, Frozen, Hold | 0 | — | — | 73.0 | |
| Eggs, Dried, whole | 35 | Low as pos. | 6 mo. | 6.0 | — |
| Eggs, Dried, yolk | 35 | Low as pos. | 6 mo. | — | — |
| Eggs, Dried spray albumen | 35 | Low as pos. | 6 mo. | Up to 6 | — |
| Eggs, Fermented albumen | Room temp. | Low as pos. | — | 3-15 | — |
| Endive (escarole) | 32 | 90-95 | 2- 3 wk. | 93.3 | 30.9 |
| Figs, Dried | 40-45 | 65-75 | 9-12 mo. | 24.0 | — |
| Figs, Fresh | 28-32 | 65-75 | 5- 7 days | 78.0 | 27.1 |
| Fish, Fresh | 33-40 | 90-95 | 5-20 days | — | — |
| Fish, Frozen | 0-10 | 90-95 | 8-10 mo. | — | — |
| Fish, Smoked | 40-50 | 50-60 | 6- 8 mo. | — | — |
| Fish, Brine Salted | 40-50 | 90-95 | 10-12 mo. | — | — |
| Fish, Mild cured | 28-35 | 75-90 | 4- 8 mo. | — | — |
| Frozen-pack fruit | −10- 0 | — | 6-12 mo. | — | — |
| Frozen-pack veg. | −10- 0 | — | 6-12 mo. | — | — |
| Garlic, dry | 32 | 70-75 | 6- 8 mo. | 74.2 | 25.4 |
| Gooseberries | 31-32 | 80-85 | 3- 4 wk. | 88.3 | 28.9 |
| Grapes, American type | 31-32 | 80-85 | 3- 8 wk. | 81.9 | 27.5 |
| Grapes, European type | 30-31 | 80-85 | 3- 6 mo. | 81.6 | 24.9 |
| Horseradish | 32 | 95-98 | 10-12 mo. | 73.4 | 26.4 |
| Kohlrabi | 32 | 95-98 | 2- 4 wk. | 90.1 | 30.0 |
| Leeks, greens | 32 | 85-90 | 1- 3 mo. | 88.2 | 29.2 |
| Lemons | 55-58 | 85-90 | 1- 4 mo. | 89.3 | 28.1 |
| Lettuce | 32 | 90-95 | 2- 3 wk. | 94.8 | 31.2 |
| Limes | 45-48 | 85-90 | 6- 8 wk. | 86.0 | 29.3 |
| Logan-blackberries | 31-32 | 80-85 | 7-10 days | 82.9 | 29.5 |
| Bacon, Fresh | −10- 0 | 90-95 | 4- 6 mo. | — | — |
| Bacon, Smoked | 60-65 | 85 | 4- 6 mo. | 13.29 | — |
| Beef, Fresh | 32-34 | 88-92 | 1- 6 wk. | 62.77 | 28-29 |
| Beef, Frozen | −10- 0 | 90-95 | 9-12 mo. | — | — |
| Fat backs | 38-40 | 85-90 | 0- 3 mo. | — | — |
| Ham shoulders, Fresh | 32-34 | 85-90 | 7-12 days | 47-54 | 28-29 |
| Ham shoulders, Frozen | −10- 0 | 90-95 | 6- 8 mo. | — | — |
| Ham shoulders, Cured | 60-65 | 75-80 | 0- 3 yr. | 40-45 | — |
| Lamb, Fresh | 32-34 | 85-90 | 5-12 days | 60-70 | 28-29 |
| Lamb, Frozen | −10- 0 | 90-95 | 8-10 mo. | — | — |
| Pork, Fresh | 32-34 | 85-90 | 3- 7 days | 35-42 | 28-29 |
| Pork, Frozen | −10- 0 | 90-95 | 4- 8 mo. | — | — |
| Sausage, Smoked | 40-45 | 85-90 | — | — | — |
| Sausage, Casings | 40-45 | 85-90 | — | — | — |
| Veal | 32-34 | 90-95 | 5-10 days | 70-80 | 28-29 |
| Melons, Casaba, Persian | 36-40 | 75-85 | 4- 6 wk. | — | — |
| Melons, Honeydew, honeyball | 36-38 | 75-85 | 2- 4 wk. | — | 29.0 |
| Melons, Muskmelons | 32-34 | 75-78 | 7-10 days | 92.7 | 29.0 |
| Melons, Watermelons | 36-40 | 75-85 | 2- 3 wk. | 92.1 | 29.2 |
| Mushrooms | 32-35 | 80-85 | 2- 3 wk. | 91.1 | 30.2 |
| Nuts, dried | 32-50 | 65-75 | 8-12 mo. | 3.6 | — |
| Oleomargarine | −10- 0 | — | — | — | — |
| Olives, fresh | 45-50 | 85-90 | 4- 6 wk. | 75.2 | 28.5 |
| Onions, onion sets | 32 | 70-75 | 6- 8 mo. | 87.5 | 30.1 |
| Oranges | 32-34 | 85-90 | 8-10 wk. | 87.2 | 28.0 |
| Parsnips | 32 | 90-95 | 2- 4 mo. | 78.6 | 28.9 |
| Peaches | 31-32 | 80-85 | 2- 4 wk. | 86.9 | 29.4 |
| Pears | 29-31 | 85-90 | — | 83.5 | — |
| Peas, green | 32 | 85-90 | 1- 2 wk. | 74.3 | 30.0 |

| Commodity | Storage Temperature, F | Relative Humidity, % | Approximate Storage Life | Water Content, % | Average Freezing Point, F |
|---|---|---|---|---|---|
| Peppers Sweet | 32 | 85-90 | 4- 6 wk. | 92.4 | 30.1 |
| Peppers Chili (dry) | 32-50 | 70-75 | 6- 9 mo. | — | — |
| Persimmons | 31-32 | 85-90 | 2- 3 wk. | 78.2 | 28.3 |
| Pineapples Mature green | 50-60 | 85-90 | 3- 4 wk. | — | 29.1 |
| Pineapples Ripe | 40-45 | 85-90 | 2- 4 wk. | 85.3 | 29.9 |
| Plums | 31-32 | 80-85 | 3- 8 wk. | 85.7 | 28.0 |
| Potatoes | 38-50 | 85-90 | — | 77.8 | 28.9 |
| Pumpkins or squash | 50-55 | 70-75 | 2- 6 mo. | 90.5 | — |
| Quinces | 31-32 | 80-85 | 2- 3 mo. | 85.3 | — |
| Radishes, winter | 32 | 95-98 | 2- 4 mo. | 93.6 | — |
| Rabbits Fresh | 32-34 | 90-95 | 1- 5 days | — | — |
| Rabbits Frozen | −10- 0 | 90-95 | 0- 6 mo. | — | — |
| Raspberries Black | 31-32 | 80-85 | 7-10 days | 80.7 | 29.9 |
| Raspberries Red | 31-32 | 80-85 | 7-10 days | 83.4 | 30.4 |
| Rhubarb | 32 | 90-95 | 2- 3 wk. | 94.9 | 28.4 |
| Rutabagas | 32 | 95-98 | 2- 4 mo. | 89.1 | 29.5 |
| Salsify | 32 | 95-98 | 2- 4 mo. | 79.1 | 28.4 |
| Spinach | 32 | 90-95 | 10-14 days | 92.7 | 30.3 |
| Strawberries | 31-32 | 80-85 | 7-10 days | 90.0 | 29.9 |
| Sweet potatoes | 55-60 | 75-80 | 4- 6 mo. | 68.5 | 28.5 |
| Tomatoes Mature green | 55-70 | 85-90 | 3- 5 wk. | 94.7 | 30.4 |
| Tomatoes Ripe | 40-50 | 85-90 | 7-10 days | 94.1 | 30.4 |
| Turnips | 32 | 95-98 | 4- 5 mo. | 90.9 | 30.5 |

Note: Although the subject is not discussed in this paper, it is worth while to note here that the proper stowage of perishables is essential if serious spoilage is to be avoided. Containers and packages should be arranged to provide ample passage for the circulation of conditioned air throughout the cargo.

*Reciprocating Compressor Performance.*

The development of quick frozen foods and the increase in its distribution has influenced the design of cargo refrigeration. Previously, refrigerated cargo compartments were designed for a minimum temperature of 0°F. Some shippers of quick frozen foods require —10°F. to be maintained. To obtain this temperature it is necessary to operate the compressor at suction temperature of approximately —25°F. for direct expansion systems or —35°F. for indirect or brine systems. At a condensing temperature of 105°F. this results in a compression ratio of 10.4 and 13.3 respectively. Freon-12 reciprocating compressors not equipped with water jackets reach the limit of practicability at a compression ratio of about 12. At higher compression ratios it is usually advisable and economical to use two compressors in series either by direct staging or by cascading. A curve of the volumetric efficiency for a typical Freon-12 reciprocating compressor is shown in the accompanying graph (Fig. 1). A comparison of the performance of a typical reciprocating Freon-12 compressor operating at various suction and condensing temperatures is presented in Table 4.

## TABLE 4: PERFORMANCE COMPARISON OF TYPICAL RECIPROCATING F-12 COMPRESSOR (LIQUID SUBCOOLED 10 DEGREES)

| Condensing Temp. F ..... | +105 | +105 | +105 | +105 |
|---|---|---|---|---|
| Suction Temp. (Corresponding to suction pressure at Compressor, F.) ..... | + 15 | — 5 | — 20 | — 35 |
| **Item** | | | | |
| Compression ratio P2 P1 ..... | 4.3 | 6.6 | 9.2 | 13.3 |
| Ref. Effect btu/lb. ..... | 49.7 | 47.6 | 46 | 44 |
| Cycle Efficiency, % ..... | 76 | 71 | 66 | 61 |
| Coef. of Perform. ..... | 4.1 | 3.2 | 2.6 | 2.1 |
| Over-all vol. eff. % ..... | 75 | 65 | 52 | 35 |
| Brake hp per ton ..... | 1.6 | 2.3 | 3.2 | 5.3 |

It is to be noted, in Figure 1 and Table 4, that the over-all volumetric efficiency falls off sharply as the compression ratio increases. It is also interesting to note that brake horsepower per ton is doubled when the suction temperature is reduced from +15° to —20° and is more than tripled when the suction temperature is reduced to —35°. It is, therefore, important to select carefully the maximum permissible temperature to be carried during the voyage.

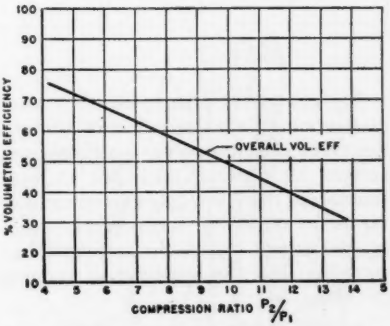



FIG. 1.—Typical volumetric efficiency curve for Freon-12 compressor.

*Centrifugal Refrigerating Machines.*

In certain applications the multi-stage centrifugal refrigerating machine has been found to have many advantages, among which are the following:

1. Occupies less space than any other type.
2. Inherently more efficient than other compressors under comparable operating conditions.
3. Has fewer wearing parts, reducing maintenance costs to a minimum.

The centrifugal refrigerating machine is available in sizes greater than 100 HP and is, therefore, limited to large installations.

Six American President Line C-3 Cargo vessels, completed shortly after the end of World War 2, are each equipped with two 4-stage Freon-11 centrifugal refrigerating machines, direct connected to 250 HP. turbines. The refrigerated space is divided among 13 compartments, each equipped with a Carrier cold diffuser and capable of maintaining any temperature from —10°F. to plus 55°F. This installation has proven highly successful, particularly in the transportation of frozen foods at —10°F.

# SOME FACTORS IN THE COMPARISON OF PROPELLING MACHINERY TYPES.

ACKNOWLEDGMENT.

In this article, reprinted from "The British Motorship" for January 1949, C. C. POUNDER, Chief Engineer of Harland and Wolff give an interesting evaluation of some important propulsion machinery design considerations. These are extracts from a paper read before the Liverpool Engineering Society on December 15, 1948.

Any assessment made by the shipowner to determine the value to him of alternative types of prime-mover must take into account all factors and in the ultimate summation there are only two criteria, so far as the shipowner is concerned, namely: the capital cost of the specified ship, complete; and the total cost of running the ship.

In everyday life the comparisons most frequently required are those between Diesel engines and steam turbine machinery. The endeavor of this paper is to point out the more important considerations which enter into any such comparisons.

In outline, the proposition—as it normally arises—is simple. A shipowner requires a ship, of stated quality, to carry a stated weight, or cubic capacity, of cargo at a stated speed. The design of the vessel is a problem for the shipbuilder, who determines the size of the ship and its first cost. The running of the ship, with all the factors and variables which enter into the matter, is a problem for the shipowner, through his superintendents and his accountants. The function of the engine manufacturer is to provide the shipbuilder with particulars of the machinery weights and spaces for the power required; to state the fuel consumption per day; and to prescribe the amount of lubricating oil, and so on, to be carried on board.

For purposes of ascertaining real machinery weight and space, it is necessary to segregate from the complete ship all those physical items which are called into being by the machinery and which are essential for its running throughout the stated voyage period.

Comprised within the compass of a true machinery comparison, and expressed in general terms, are: the dry weight of all the machinery in sea-going condition; the weight of fuel to be carried for the required "steaming" radius; the space occupied by the propelling machinery; the space occupied by the fuel bunkers; the weight of steel needed for the bunkers; the weight and space occupied by the lubricating oil, water and so on. The effect of appreciable differences in these and kindred items will be to alter the ship dimensions and cost for any given cargo deadweight or cubic capacity. Differences in efficiency will be shown by the respective amounts of fuel needed for specific powers—as

determined by speed of ship under given conditions of loading—for any prescribed distance. If the engine revolutions are not the same for the machinery types compared, the difference in propulsive efficiency must be taken into account, as this affects the weight of fuel burned to propel the ship at given speed. There are other matters, also, to be considered. Thus, if the quality of fuel is not the same, differences in specific cost must be allowed for. The price to be paid for lubricating oil must not be lost sight of. Then there is the cost of labor to work the machinery at sea and in harbor; the cost of overhaul for surveys and maintenance; and the effect, upon the turnround of the vessel, of the time needed in port for overhauling work. Finally, there are such points as insurance differentials, harbor dues, and so on, which arise when the compared vessels, equal in cargo-carrying capacity, differ in first cost and tonnage.

It may be useful to apply the principles, implied in the foregoing, to two typical examples. Attention will be confined to matters within the purview of the builder.

*7,500 s.h.p.; Diesel and Steam Machinery Compared.*

The first example is a single-screw installation of 7,500 s.h.p. and the comparison is between two-stroke, single-acting, Diesel machinery and high-pressure-temperature steam turbines. The ship is a cargo-liner, with engines and amidships.

(a) *Weights*

The basis of comparsion is the weight of the Diesel engine installation.

*Diesel Engine Scheme.*—The propelling engine is a single-acting, two-stroke, crosshead type unit, having seven cylinders 750 mm. (29.53 ins.) bore 2,000 mm. (78.74 ins.) combined main and exhaust piston stroke. Fig. 1 shows the engine in section. This is a standard size of engine, as exemplified by the numbers being constructed for British, Dominions and Continental shipowners. It delivers 7,500 s.h.p. at 105 r.p.m., the mean piston speed being 5.25 metres per sec. (1,033 ft. per min.), the indicated mean pressure 6.4 kg. sq. cm. (91.0 lb. sq. in.). Assuming a very comprehensive machinery specification, the make-up is: weight of main engine over crankends, including thrust block and shaft, gratings, ladders, all pipes about the engine, also spare gear = 576 tons; all tunnel shafting, sterntube, propeller, plummer blocks, also spares—including bronze propeller and tailshaft = 131 tons; auxiliary Diesel engines, all

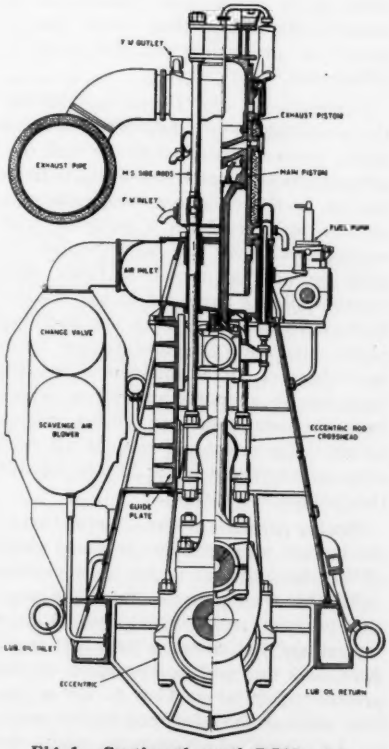



**Fig. 1.—Section through 7,500 s.h.p. seven-cylinder Harland-B. and W. single-acting engine.**

pumps in machinery space, auxiliary boilers, air reservoirs, also all attendant spare gear = 183 tons; all pipe arrangement work, including floors, gratings, ladders, tanks, silencers, funnel, workshop, storerooms and contents, ventilation, etc. = 173 tons; fresh water in jacket circulating system, including water in make-up and drain tanks, salt water in coolers and circulating system, lubricating oil in circuit and in drain tanks, also including spare charge of lubricating oil = 40 tons; i.e., total weight of machinery in sea-going condition = 1,103 tons.

In the above: the crankshaft is 10 per cent. in strength above Lloyd's; the line shafting 5 per cent., and the tailshaft 10 per cent. The shafting is assumed to be 180 ft. long, reckoned from thrust shaft coupling to tailshaft thimble-end.

From this point, for the remaining items in the comparison, a great deal of monotonous arithmetical detail can be saved by proceeding on the basis of differences.

*Seatings.*—In the Diesel installation, the main engine is bolted-down to thick plates riveted or welded to the tank-top, intercostals being provided in the double-bottom. In the turbine arrangement, the turbine is carried on a built-up steel seating, and there are seatings also for gearcase and main boilers. Here, also, intercostals are provided in the double-bottom. In addition, there are doubling plates and ship attachments for feed-water heaters, forced and induced draught fans, uptake supports and so on, which have no counterpart in Diesel engine weights. The algebraic sum of all such items and contra-items is 33 tons against the turbine installation.

*Bunker capacities.*—An examination of the bunker arrangements in many ships of the size and type under consideration —Atlantic routes apart—shows the capacity to vary from 35 days to 60 days "steaming." A common capacity is 45 days, and this will be the basis of the present comparison. That is, for a turbine steamer the additional bunker space to be provided—compared with the Diesel ship—is the difference in fuel consumption between the two prime-movers. Assuming, for overall purposes, a fuel consumption rate of 0.41 lb. per s.h.p. hour for the Diesel machinery and 0.61 lb. for the steam turbine, the additional bunker weight for the latter, for 45 days, is 720 tons. The port consumption is assumed to be the same for both types of machinery. The additional bunker weight and space can only be obtained by increasing the ship dimensions, always assuming the same cargo-deadweight and cubic capacity.

*Reserve feed-water tanks.*—For a high-pressure-temperature steam turbine installation, it is necessary to carry—usually in the double-bottom—an appreciable amount of fresh water for reserve feed purposes. In addition, one or more independent distilled water tanks, inside the machinery spaces, are provided for make-up feed purposes. These items have no counterpart in a Diesel engine installation.

Examination of characteristic ships shows the minimum amount of reserve feed water to be 150 tons, with 25 tons in two distilled water tanks, i.e., a total of 175 tons. The owner decides upon the amount of reserve feed water to be carried on board.

*Steel weights of bunkers and other items.*—Usually, the double-bottom tanks are completely made use of for the carriage of fuel oil. That is, the total bunker weight is the sum of the fuel carried in the double-bottom tanks plus that stored in bunker deep tanks. Fig. 2 shows a typical longitudinal section of an actual single-screw ship, propelled by a 7,500 s.h.p. Diesel engine. It will be seen that all available double-bottom tanks are allocated the fuel oil. This ship is bunkered for 50 days. It follows from the foregoing that, for a steamer, the additional bunkers can only be obtained by an extension of the deep tanks. All other things being the same, this means an increase in the dimensions of the ship, which means an increase in the steel weight.

The storing of reserve feed water in the double-bottom means, for the steamer, the displacement of an equivalent amount of fuel oil. This amount must, therefore,

be added to the bunker deep tanks, further to increase their size. It should be noted that bunkers for the carriage of fuel oil are necessarily of heavy construction.

Increase in bunker dimensions can mean greater surfaces for the heating grids, longer lengths for the suction, delivery and vapor piping; and so on.

The author's experience has not borne out this notion. The propeller diameter and its revolutions are usually determined by the stern-end design of the ship, irrespective of the nature of the prime-mover.

So far as the immediate comparisons are concerned, the revolutions are assumed to be the same for steamer and

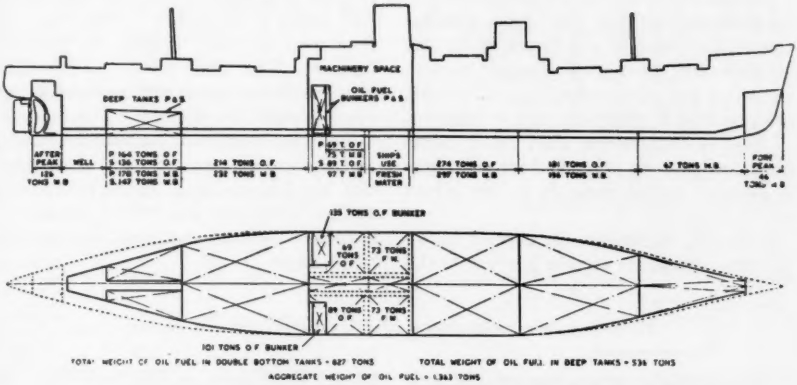

MACHINERY SPACE
DEEP TANKS P & S
OIL FUEL BUNKERS P & S
AFTER PEAK
WELL
SHIPS USE
FRESH WATER
FORE PEAK
135 TONS O.F BUNKER
69 TONS O.F
73 TONS F.W.
101 TONS O.F BUNKER
TOTAL WEIGHT OF OIL FUEL IN DOUBLE BOTTOM TANKS = 827 TONS
TOTAL WEIGHT OF OIL FUEL IN DEEP TANKS = 536 TONS
AGGREGATE WEIGHT OF OIL FUEL = 1,363 TONS

**Fig. 2.—Arrangement of fuel oil tanks for a 7,500 b.h.p. installation.**

The weight requirements of the steamer cannot be met—on an equal basis—without increasing the size of the ship. This slightly heavier ship requires slightly more power, which adds a little to the machinery weight, which fractionally increases the fuel consumption, and so on. Differences in engine casing weights and a number of other things are also to be allowed for. Pursuit of these matters to their conclusion shows the weight of steel and other materials in the steamer to be 232 tons more than in the Diesel-engined ship.

*Propeller revolutions.*—It is commonly said that one of the advantages of the geared turbine is that the revolutions at the turbine, and also at the propeller, can be so chosen that the highest respective efficiencies are ensured both for turbine and propeller, whereas the Diesel engine must necessarily always be a compromise —with its propeller rotating faster than the turbine propeller—because of the need to keep down the size, weight and cost of the Diesel engine.

Diesel ship. Hence no adjustment to the propulsive efficiency is needed.

*Steam machinery weight.*—With the same level of practice as for the Diesel machinery, and with an equally comprehensive specification, the records available to the author show as follows: All tunnel shafting items, all pumps, turbo-generators, auxiliary boiler, with all attendant spare gear; all pipe arrangement work; water and oil in systems: aggregate weight equals 518 tons.

The above excludes only the main boilers, main turbine and main condenser. With a boiler working pressure of 450 lb. sq. in., by gauge, at the superheater outlet and a temperature of 800 degrees F., the aggregate weight for these items, at 105 r.p.m., can be assumed to be 240 tons—this weight being similar to that recently given by steam experts for their very latest proposals. It is here accepted uncritically. The turbine is a single cylinder unit.

The total weight of steam machinery in sea-going condition is thus 758 tons.

*Comparison of Diesel and steam weights.*—According to the above analysis the comparative position is now as under, viz:—

| | |
|---|---|
| Weight of Diesel machinery in sea-going condition... | 1,103 tons. |
| Weight of steam machinery in sea-going condition... | 758 tons. |
| Balance in favor of steam machinery..... | 345 tons. |

Contra-items, which are differences between the Diesel and the steam installations, and which are adverse to the steamer are: seatings = 33 tons; bunker, etc., steel and other materials = 232 tons; reserve feed = 175 tons; total = 440 tons.

That is, before any fuel oil is taken into the bunkers, the machinery weight is adverse to the steamer to the extent of 95 tons.

Assuming bunkers to be carried for 45 days the aggregate weight adverse to the turbine installation = 815 tons. For any other number of days the change will be proportionate. Similarly, for the mean voyage weights the necessary adjustments can easily be made.

If the double-bottom of the Diesel-engined ship is made abnormally deep, with a shallow bedplate arranged to suit, the adverse steel weight will be reduced. The deeper double-bottom tanks thus created will carry more oil fuel, which will mean a reduction in the size and therefore in the weight of the side bunkers. The total effect of all these changes is to reduce the amount of adverse weight by 40 tons.

The real weight of turbine machinery and Diesel engines, of equivalent practice, can become equal at the middle of a voyage when the size of the bunkers is reduced to suit a "steaming" radius of 14 to 16 days.

*An alternative comparison of Diesel and steam weights.*

An optimistic steam man might well say that he could produce a high-pressure-temperature installation, of the power under discussion, for 120 r.p.m., with a much lower weight than that mentioned above. He might give figures as under, viz: boilers (no water), turbine, gearing and condenser = 236 tons; shafting, stern tube, propeller, etc. = 75 tons; auxiliaries, including turbo-generators = 57 tons; pipes and fittings = 103 tons; funnel, ladders, gratings, floorings, tanks and spare gear = 89 tons; water in boilers, oil and water in systems (excluding reserve oil) = 32 tons; total weight of machinery in seagoing condition = 592 tons. (Vide Trans. I.Mar. E. Vol. LIX p. 255.)

These figures obviously are for a specification different from that quoted earlier for the Diesel installation. If, therefore, such a specification is acceptable to a shipowner for steam plant, an equivalent specification for main engine and all ancillary items should be acceptable for Diesel machinery. Notwithstanding this, on grounds of prudence the author would not like to suggest a total Diesel machinery weight less than 933 tons.

Other adjustments may be made. The reserve feed water, plus independent distilled water tanks, may be reduced to 115 tons. The turbine propeller revolutions are 120 per minute, against 105 r.p.m. for the Diesel engine. For the type of ship under consideration, this means 3 per cent. of the propulsive efficiency adverse to the turbine. The effect when pursued to its end is an increase in the adverse weight of steel for bunkers, seatings, etc., from 265 to 271 tons. Water and oil at 32 tons for the turbine machinery does not line-up with 40 tons for the Diesel installation. However, as there is an allowance of 15 tons for a spare oil charge in the Diesel list, which does not appear in the turbine plant, the extent of the adjustment will be restricted to this amount. In the previous comparison this item was ignored.

The comparison of weights thus becomes:

| | |
|---|---|
| Weight of Diesel machinery in sea-going condition.... | 933 tons. |
| Weight of steam machinery in sea-going condition.... | 592 tons. |

Balance in favor of steam.. 341 tons.

Contra-items adverse to the steam proposal are: bunker, seatings, etc., steel and other materials = 271 tons; reserve feed = 115 tons; lubricating oil = 15 tons; total = 401 tons.

That is, in "steam-up" condition, the weight of machinery is 60 tons adverse to the turbine. With bunkers for 45 days, the aggregate weight adverse to the turbine proposal is 780 tons. Table I summarizes the comparisons.

**Table I.—7,500 s.h.p. installation**

Weight differentials (tons): direct-coupled Diesel and turbine machinery

| | Direct-coupled Diesel | Turbine | Alternative direct-coupled Diesel | Alternative turbine |
|---|---|---|---|---|
| Weight of machinery in sea-going condition | 1,103 | 758 | 933 | 592 |
| Weight of reserve feed water | — | 175 | — | 115 |
| Differential of above weights | 170 (adverse) | — | 226 (adverse) | — |
| Differential weight of lubricating oil | — | — | — | 15 (adverse) |
| Differential weight of steel in bunkers, etc. | — | 255 (adverse) | — | 271 (adverse) |
| Differential at beginning of voyage (without bunkers) | — | 95 (adverse) | — | 60 (adverse) |
| Differential in bunkers for 45 days | — | 720 (adverse) | — | 720 (adverse) |
| Total differential at beginning of voyage, including bunkers | — | 815 (adverse) | — | 780 (adverse) |

*Geared Diesel engines.*—As an alternative to the direct-coupled Diesel engine, it is instructive to consider the potentialities of geared Diesel engines.

Assuming the same level of practice, together with the same comprehensiveness of specification, as for the direct-coupled installation, the weight of a geared Diesel engine plant in sea-going condition is 922 tons.

The engine unit is illustrated in Fig. 3. There are over 40 engines of this type

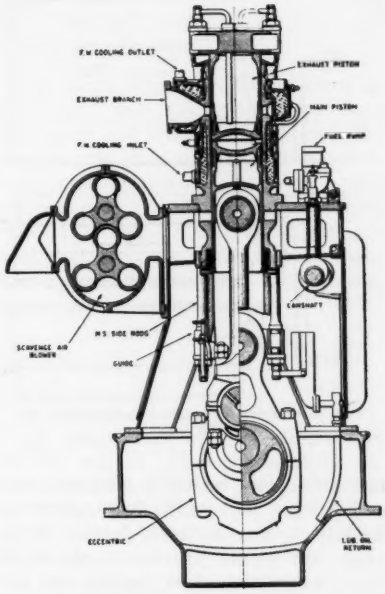



**Fig. 3.—Seven-cylinder 3,750 b.h.p. engine for a two-engine geared installation.**

and size under construction at present. Each of the two engines is a seven-cylinder, single-acting, two-stroke, trunk unit, 530 mm. (20.87 in.) bore, 1,180 mm. (46.46 in.) total stroke, 220 r.p.m., ratio of revolutions 2.2:1. The mean piston speed is 6.0 metres per sec. (1,180 ft. per min.), brake mean effective pressure 4.5 kg. sq. cm. (64.0 lb. sq. in.). It is a quiet, steady-running engine. It will run smoothly down to 40 r.p.m. There are two engines, arranged as shown diagrammatically in Fig. 4. The engines transmit their power to the tunnel shafting and propeller through electric slip-couplings and single-reduction gearing. The weight build-up is as follows, viz: weight of main engines, gearing, magnetic couplings, thrust blocks, gratings, pipes, etc., around engines, including spare gear to Lloyd's requirements = 377 tons; shafting, stern tube, propeller, plummer blocks, etc., also spares, including bronze propeller and tail shaft, 133 tons; auxiliary engines, pumps, auxiliary boiler, air

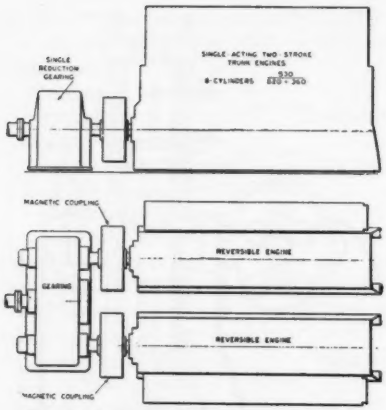



**Fig. 4.—Layout of geared installation of 7,500 b.h.p.**

reservoirs, etc., including all spare gear = 195 tons; pipe work, floors, gratings, ladders, tanks, silencers, funnel, workshop, storerooms, ventilation, etc. = 177 tons; water and oil in engines and circuits = 40 tons; total = 922 tons.

The engine bedplates are of the flat-bottomed design, as indicated in Fig. 3.

The weight of 922 tons is comparable with the turbine machinery weight of 758 tons.

Although, by using a suitable propeller, the geared Diesel engine fuel consumption can sometimes be made as low as that of the direct-coupled engine, for the purposes of this comparison the consumption may be taken as 10 per cent. greater than that of the direct-coupled engine—say 0.45 lb. s.h.p. hour, all purposes. Against this there is a credit, to the geared engine, of 4 per cent. for its lower propeller revolutions, which reduces the fuel rate to 0.43 lb. s.h.p. hour. The items adverse to the turbine are: seatings, bunker, etc., steel and other materials = 241 tons and reserve feed = 175 tons.

The balance sheet therefore stands thus:

| | |
|---|---|
| Weight of steam machinery, sea-going condition ..... | 758 tons. |
| Differential items, adverse to steam machinery..... | 416 tons. |
| Total | 1,174 tons. |
| Weight of geared Diesel machinery, sea-going condition ..... | 992 tons. |
| Machinery weight adverse to steam ..... | 252 tons. |

*An alternative geared-engine comparison.*—If a specification equivalent to the turbine machinery weight of 592 tons is accepted, the general Diesel weight becomes 797 tons. With contra-items of 362 tons, the "steam-up" machinery condition, excluding bunkers, is 157 tons adverse to steam machinery). (See Table II.)

**Table II.—7,500 s.h.p. installation**

Weight differentials (tons): geared Diesel and turbine machinery

| | Geared Diesel | Turbine | Alternative geared Diesel | Alternative turbine |
|---|---|---|---|---|
| Weight of machinery in sea-going condition | 922 | 758 | 797 | 592 |
| Weight of reserve feed water | — | 175 | — | 115 |
| Differential of above weights | — | 11 (adverse) | 90 (adverse) | — |
| Differential weight of lubricating oil | — | — | — | 15 (adverse) |
| Differential weight of steel in bunkers, etc. | — | 241 (adverse) | — | 232 (adverse) |
| Differential at beginning of voyage (without bunkers) | — | 252 (adverse) | — | 157 (adverse) |
| Differential in bunkers for 45 days | — | 648 (adverse) | — | 648 (adverse) |
| Total differential at beginning of voyage, including bunkers | — | 900 (adverse) | — | 805 (adverse) |

*Other prime-movers.*—There is not space to make a detailed comparison with turbo-electric or Diesel-electric installations. It must suffice to state that, for similar specifications, the weights exceed those for geared turbines. The fuel consumption for turbo-electric sets is higher than for geared turbines; and that of Diesel-electric plant greater than for direct-coupled Diesel engines.

The total weight of turbo-electric machinery, plus bunkers for 45 days, is approximately 1,100 tons greater than for direct-coupled Diesel engines, depending upon the choice of details; for Diesel-electric machinery it can be approximately the same as for direct-coupled engines.

(*b*) *Space Occupied.*

The machinery space volume is still governed by the Merchant Shipping Act of 1894 (Part 1, Section 78, Clause 1 (a)), which states that in screw vessels, in which the tonnage of the space solely occupied by and necessary for the proper working of the machinery is above 13 per cent. and under 20 per cent. of the gross tonnage the deduction allowance for the purpose of register tonnage shall be 32/100 of the gross tonnage. The Act stipulates that, outside these limits, the tonnage deduction allowance shall be the actual tonnage of the machinery space plus three-quarters of that space. Thus, if the machinery space were 10 per cent. of the gross tonnage, the deduction allowance would be only 17.5 per cent. instead of the 32 per cent. deduction allowed for 13 per cent.

As the engine-rooms of Diesel-engined ships are usually on the 13 per cent. border-line—sometimes it is necessary to make upward adjustments to ensure the 13 per cent.—there is no point in discussing what might or might not be achieved with engine-room spaces, because shipowners will not forego the advantage to them of the reduction in tonnage dues obtainable by observance of the 13 per cent. rule.

The bunkers for 45 days require, for the turbine ship, an additional volume of 27,500 cubic ft., compared with the direct-coupled Diesel installation.

(*c*) *First Cost.*

For equivalent specifications and with everything in equity, the first cost of a high-pressure-temperature turbine installation is usually higher than that of a direct-driven Diesel plant; a common figure is 6 per cent. to 7 per cent., but sometimes it is more.

This is not surprising. Turbine installations, with their ever-growing ancillary equipment and with the increasingly expensive materials required, tend to become more elaborate and more costly, whereas, with Diesel machinery the trend is towards simpler designs and the almost exclusive use of mild steel and cast iron. The hull of a Diesel-engined ship, being smaller, is also cheaper.

(*d*) *Fuel Prices.*

Oil fuel prices bear no relation to calorfic values. The continuously increasing world demand for lighter oils is having the effect of de-grading the quality of the fuel oils available for marine use.

If the price ratio of Diesel oil to boiler oil exceeds about 1.5:1, the oil fuel bill becomes adverse to the Diesel engine. The average ratio at present seems to be about 1.2:1. The Diesel engine is tending towards the use of lower grade oils.

The cost of 35 to 40 gallons of lubricating oil per day for the Diesel machinery must be taken into account.

(*e*) *Maintenance Costs.*

During—and since—the war, overhauling costs have been, on the whole, against the Diesel engine.

Routine lying-up periods excepted, the propelling machinery, whatever the type may be, should be capable of being dealt with during the time available for cargo handling.

Looking to the future, the position may be different. The wealth of experience now gained with the later and more advanced types of Diesel engine implies a quicker turn-round for Diesel machinery. From now onwards the attractiveness of steam machinery, on this count, can be expected to decline.

*13,000 s.h.p.; Diesel and Steam Machinery Compared.*

By far the greater number of ships of this power, whether Diesel driven or steam driven, are required by their owners, for divers reasons, to have twin screws.

A comparison of twin-screw Diesel machinery and steam turbines follows the pattern indicated for the 7,500 s.h.p. single-screw ship. The results, as regards weight of machinery and space occupied, show the same order of differences.

At the present time, in the different shipbuilding centers — Belfast amongst them—a number of turbine-driven vessels are being built, which deliver 13,000 s.h.p. through a single-screw. Single-screw ships of this power are, as already implied, very much the exception.

A single-screw, direct-coupled Diesel engine of well-tried design, having moderate rating and low revolutions, will show an overall result favorable to the Diesel engine, when compared with a single-screw turbine. Such an engine would be of the size and type shown in Fig. 5, the cylinders 660 mm. (25.98 ins.) bore, 1,500 mm. (59.06 ins.) stroke. But the author would not normally offer such a proposition. This does not, however, leave the turbine with a clear field for single-screw ships of this power. The natural competitor of the large, single-screw, geared steam turbine is the single-screw geared Diesel engine.

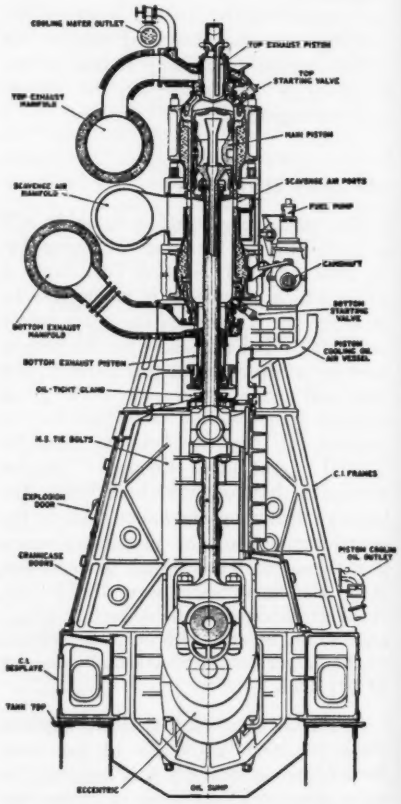



**Fig. 5.—Section through cylinder of 13,000 s.h.p. Harland double-acting engine.**

A very suitable arrangement consists of four single-acting, two-stroke, trunk-type engines, transmitting their power through electric slip-couplings and single-reduction gearing to the propeller shafting. The engine units are 530 mm. (20.87 ins.) bore, 820 mm. (32.28 ins.) main stroke, 360 mm. (14.17 ins.) exhaust stroke, six cylinders per engine, 220 r.p.m. piston speed 6.0 metres per sec. (1,180 ft. per min.), brake mean effective pressure 4.6 kg. sq. cm. (65.4 lb. sq. in.), 100 r.p.m. preferred for the propeller—if the sternend design permits—otherwise as near this as practicable—say 105 r.p.m.

*Miscellaneous Comments.*

The Diesel engine fuel consumption figures given in the paper are averages for many engines which have been in service for many years. At the end of 20 years of service the fuel consumption of a Diesel engine is the same as at the beginning.

For a turbine installation, the fuel economy is at its best when the machinery is operating at full designed power;

experience shows that the lowest fuel consumption for a voyage is obtained when, from port to port, the run is made at continuous full power. If, for one reason or another reduced power for long periods is necessary, the fuel consumption increases appreciably. With Diesel engines the fuel rate does not vary more than 1 per cent between full power and less than half-power.

One of the things which is of real importance in arranging engine rooms is the system to be adopted for overhauling. It applies equally to all machinery types. With turbines provision may be made for withdrawing the main gearwheel up the engine casing and through the skylight. Alternatively, in the unlikely event of a wheel being withdrawn, lifting openings can be cut in all intervening decks.

In all the foregoing, comparisons for direct-coupled engines have been restricted to single-acting Diesel engines, this being the form least favorable to the Diesel engine side of the comparisons.

For double-acting engines, the only elements to be added to the single-acting engine mechanism are: the bottom cylinder covers, or exhaust pistons; fuel valves; stuffing boxes; fuel-injection equipment; and so on. There is also the addition of a bottom end to the piston, which somewhat increases the length of the cylinder liner and the piston rod. It is not generally realized that the crankshaft of a double-acter is about 28 per cent lighter than the corresponding single-acting engine crankshaft. The effect of the above-mentioned modifications is to decrease the engine weight, space and cost for the same power output. The double-acting engine makes more economical use of the structural material than does the single-actor. Fig. 6 is a section through the latest form of Harland and Wolff two-stroke, double-acting engine.

Perhaps sufficient has been written to demonstrate that the differences between a ship propelled by direct-coupled Diesel engines of moderate rating and by the latest types of high-pressure-temperature turbine are not altogether trivial. Often

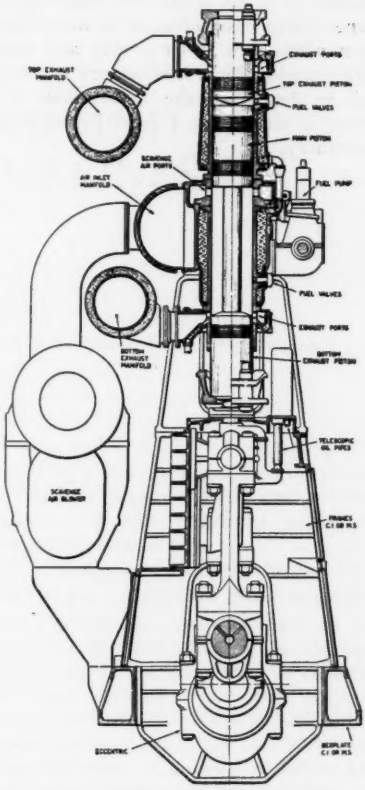


**Fig. 6.—Section through cylinder of latest type of Harland double-acting engine.**

it means five per cent and more to the capital cost of a ship.

The author has no reason to sway the figures in favor of one kind of engine at the expense of another; quite the contrary, in fact. He is responsible for the design of all the machinery built in Belfast and Glasgow by Harland and Wolff, Ltd. In 1947, when 184,000 tons of shipping were launched by the firm, of the total marine machinery output, 45 per cent consisted of steam turbines and 55 per cent of Diesel engines.

The ultimate judge of the economics of the turbine-propelled vessel or the Diesel-engined ship for any particular trade must necessarily be the shipowner

himself. It is fortunate that many shipowners have clear ideas as to what they want when they order a ship and that only occasionally is it necessary to work out, in detail, all the implications involved in alternative forms of propelling machinery.

In conclusion, whatever may be the precise details involved in a comparison of prime-mover types, it is clear that to neglect everything except weight of machinery and specific fuel consumption, as is so often done, is misleading.

# PROSPECTS OF LAND AND MARINE GAS TURBINES.

ACKNOWLEDGMENT.

The following pertinent editorial comment is reprinted from "The Engineer" for 17 December, 1948.

The lectures on the gas turbine sponsored and published by the Institution of Mechanical Engineers three years ago have been widely acclaimed, both in this country and abroad, as an authoritative and comprehensive survey of the state of theoretical and practical knowledge in the subject at that time. Since then research has been continued, not with the same intensity as characterized the closing years of war, but actively enough to have produced some noteworthy experimental projects and many more promising lines of development. In particular, there has been progress towards the use of the internal-combustion turbine for purposes now predominantly served by steam or Diesel power; and if that progress appears slow compared with the phenomenal way in which the gas turbine revolutionized aircraft power plant, it must be borne in mind that the conditions are vastly different. Under the urge of war and subsequent Service requirements, every consideration was secondary to the facts, once established, that the jet engine was peculiarly suitable for aircraft and a means of significantly increasing their speed of flight. Power for industrial transport and marine service, on the other hand, is closely governed by the need for economy. It offers the competitive factor of already functioning, not unsatisfactorily, on indigenous solid or low-grade liquid fuels, and has all the inertia of a heavily capitalized, intricate and enduring organization that admits change only after rigorous proof of prospective benefit.

If, in these circumstances, the development of gas turbines for other purposes than aircraft becomes rather an act of faith than the fulfilment of an obvious specific need, the enterprize of individuals and corporate bodies who are designing and experimenting towards that end is perhaps the more commendable, while engineers in general should be no less keenly interested in the likelihood of its attainment. Most appropriately, therefore, the prospects of land and marine gas turbines was the subject of the first of a new series of six lectures on the internal-combustion turbine, again sponsored by the Institution of Mechanical Engineers, which were given by members of the staff of the National Gas Turbine Establishment on November 3 and 4. Equally appropriate, in view of the knowledge and judgment needed to appraise so conjectural a matter fairly and constructively, was the presentation of that lecture by Mr. Hayne Constant, F.R.S., the newly-appointed Director of the Establishment, who has been intimately connected with gas-turbine research and development for many years.

It was to be expected that Mr. Constant would adhere to his previously expressed belief in the superior possibilities

of the axial, as against the centrifugal, type of compressor, and would endorse the widely accepted view that large gas-turbine plants, to be more effective than equivalent steam power, must be of the closed-circuit type in which the working fluid is indirectly heated, provision being made by various auxiliary devices to abstract as much heat as possible from the products of combustion. Less commonly realized, however, is the formidable nature of the problems entailed in the design of a satisfactory heat interchanger. The great bulk, and not very good performance, of static exchangers embodying encased banks of tubes have encouraged the alternative idea of a capacity exchanger in which a matrix of highly-conducting wire is rapidly exposed by rotation, first to hot and then to colder gas, transferring heat in the process. The principle of the porous metal regenerator, which dates back to early air-engine days, has been revived recently in a device for which remarkable results are claimed, so that its application in rotary form to the closed-cycle gas turbine seems promising enough to warrant the work still needed to provide satisfactory glands for it.

The simpler open-cycle type of gas turbine also presents some problems in land applications that are as yet imperfectly solved. Prominent among them is that of preventing dust from eroding, or being deposited on, the blades and passages of the compressor and the turbine, thereby spoiling their shape and impairing their aerodynamic efficiency. There is need, therefore, for means of filtering particles as fine as smoke out of the air entering the compressor, and of achieving the combustion of relatively cheap residual oils, not only at the efficiencies approaching 100 per cent already in sight, but also without entraining ash particles in the hot gases passing to the turbine. These factors are especially intractable in the case of the railway locomotive because of the rigid limitations upon its transverse dimensions and the variable and polluted atmospheres which it encounters in service. For some time, nevertheless the bulk and weight inherent in the closed-cycle gas turbine with heat exchanger are prohibitive, and the simpler types of gas-turbine locomotive under construction can be justified by the value of the experience to be gained with them. In Mr. Constant's opinion, however, they will compare unfavorably with equivalent Diesel-electric locomotives. He believes that eventual superiority will depend on the combination of a closed-cycle oil-burning turbine incorporating a rotary capacity-type exchanger, with wholly mechanical transmission of power to the wheels. Such a locomotive, weighing 80 tons for 3,000 h.p., having a 5:1 pressure-ratio axial compressor and an upper gas temperature of 800 deg. C. (1,470 deg. F.), would, he estimates, realize an overall efficiency, at the axles, of 26 per cent, and surpass the Diesel-electric arrangement in respect of life and reliability.

In strongly advocating liquid rather than solid fuel for gas-turbine locomotives, Mr. Constant may be suspected, perhaps, of appreciating better than most engineers in this country the difficulties of utilizing pulverized coal, with which the investigators of Bituminous Coal Research, Incorporated, are still struggling in the development of American projects. It is easy to agree, however, that oil lends itself far more readily than coal to stationary as well as mobile gas turbines; and it is interesting to hear, therefore, that, while considerations of political economy or national security may dictate the burning of powdered coal in gas-turbine plants for electric power generation, there is a possible balance in favor of the alternative of converting the coal into producer gas and feeding that to gas turbines. In circumstances where almost the whole of the heat in the turbine exhaust can be profitably utilized for district heating or low-pressure steam turbines, the gas-burning internal-combustion turbine promises a substantial reduction of generating costs below those of equivalent steam-turbine plant, while offering further advantages in such directions as cheaper sites and buildings for power stations. It is envisaged that half a dozen power-station

gas-turbines may be at work in this country within the next three years, and that small stand-by sets will be a popular addition to many existing stations.

The fact that the gas-turbine shows little prospect of giving its most effective service at capacities below about 1,500 h.p. limits its range of general utility on land, and while there are numerous tasks—in the metallurgical and processing industries, for example—for which it appears peculiarly well suited, the great variety of conditions and qualifications to be considered were beyond the scope of Mr. Constant's lecture. Lacking any such advantages over the small internal-combustion engine, the gas turbine has no foreseeable future in road transport. For marine propulsion, of course, the minimum power limitation is of less importance, and it would seem, from the volume of development work already in active progress and mentioned in contemporary literature, that the prospects of the internal-combustion turbine at sea are probably regarded as beter than on land. Mr. Constant thinks, however, that the relative merits for marine use of the gas turbine and the Diesel engine are not likely to be settled for a long time yet. The optimum type of turbine cycle is a particularly vexed question, while he goes so far as to assert that successful competition against efficient modern marine power of existing types turns on the ability of the internal-combustion turbine to burn heavy residual oils. When this point has been established, the marine gas turbine is expected to be as efficient as an equivalent Diesel plant, with less weight.

If one implication of Mr. Constant's review stands out more prominently than others, it is that the diversity of performance characteristics demanded by power plants in different types of industrial and marine service calls for corresponding attention to the elements of internal-combustion turbines if the latter are to show worthwhile economics. Versatility in the type of fuel is also clearly of great importance, not solely in order that the turbine shall be able to burn cheap fuels efficiently and without prohibitive maintenance, but also so that it shall be able to accommodate itself, for some years at least, to the existing enormously complex system of fuel production and distribution. Appropriate developments must, consequently, prove lengthy, while a substantial proportion of their cost will have to be borne by the industries which expect to benefit from them. For these reasons, the progress of the gas turbine on land and sea must inevitably be slower than that of the aircraft turbine, which, besides being in some respects simpler, was the subject of intense and subsidized research. Mr. Constant's suggestion, that the development of the internal-combustion turbine would be assisted by the standardization of its principal components, is interesting in so far as it derives from the feeling that an essentially pneumatic system is readily adaptable to such treatment, but perhaps it demands a longer view than can be seen clearly at the moment. To a limited extent, standardization of components and auxiliaries is practised with advantage in most of the power applications that hold out good prospects for the gas turbine; but, while it may eventually prove feasible for separate firms to concentrate on the manufacture of combustors, compressors, turbine units and heat exchangers, over a range of standard sizes, while others specialize in ducting, assembly and installation, such a division of labor is hardly to be expected until the gas turbine is beyond the development stage. So far as anyone can see at present, the industrial gas turbine will evolve gradually during the next quarter of a century, and will be most unlikely to supersede existing types of land and marine prime mover in any revolutionary fashion.

# GRINDING FINE-PITCH GEARS.

ACKNOWLEDGMENT.

This article by A. S. BEAM of the Vinco Corp., is reprinted from "Machinery" for December 1948. The article is a paper which was presented at a Symposium in Gear Tooth Grinding before the recent semi-annual meeting of the American Gear Manufacturers Association.

The process of grinding gear teeth is usually employed as a finishing operation. In the case of fine-pitch gears, however, it also serves, for the greater portion of the fine-pitch range, as a combind cutting and finishing operation.

There are two principal methods of grinding gear teeth, which may be distinguished as form-grinding and grinding by generation. Grinding is used as a finishing operation on heat-treated, hard gearing and master gears, as well as in cases where a high degree of surface smoothness is required. In some instances, grinding by generation is actually an economy over other methods of cutting gear teeth. In all cases, it is the experience of this company that grinding is practical on a wide variety of materials, such as tool steel, mild steel, stainless steel, iron, aluminum, bronze, brass, Micarta, and other compounded materials. Grinding of coarse-pitch gears is frequently confined to the tooth sides, while with fine-pitch gears, the grinding wheel usually cuts not only the sides, but also the root.

The two methods—form-grinding and generation by grinding—have certain distinguishing limitations and characteristics. The process of form-grinding employs a wheel that is shaped according to the contour of the tooth space. Truing, or dressing, of the wheel is usually performed by diamonds, but crushing can also be employed. Diamond dressing is popular because it is fast, cheap, flexible, and contributes to a high degree of accuracy. Flexibility, with respect to special pitches and pressure angles, and even modified profiles, is an important advantage of this method.

The accuracy of the form on the wheel is a prime consideration; however, it must be stated that associated problems are far less critical in the case of fine pitches than coarse pitches. Although the involute profile is a line with continuously changing curvature, it is possible, in many cases, to substitute a simple arc in place of the involute form. Since the deviations of this arc from the theoretical profile are proportional to the pitch, it is obvious that deviations which may be serious in the case of coarse pitches will be less significant in the finer pitches.

Form-grinding is particularly suitable when the number of teeth is high. Low numbers of teeth may require some under-cut, which the form-grinding process does not provide without resorting to impractical procedures. However, the thickness of material removed by undercutting, measured from the theoretical involute form, is relatively insignificant unless the required under-cut is too radical. It is possible to tolerate a slight degree of involute interference as long as this interference is less than the minimum backlash.

The process of form-grinding is sometimes associated with the misconception that liberal tooth-thickness tolerances cannot be allowed, because down feeding of the wheel results in improper profile deviations. This problem is met by keeping the position of the grinding wheel undisturbed once a satisfactory tooth profile has been established.

Changes in tooth thickness of the workpiece are then effected by a circumferential adjustment of the indexing assembly.

With this method, the grinding wheel is thinner than the space to be cut and contacts one side only. The work is finished by first indexing completely around and then repeating the process after the unfinished side has been rolled against the wheel by circumferential index readjustment, as seen at the left in Fig. 1. This assures high accuracy of

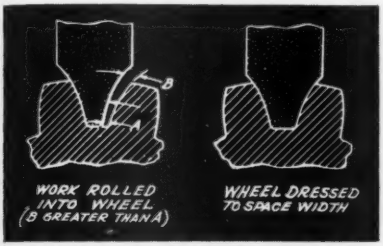



Fig. 1. Two Methods of Form-grinding Gear Teeth. At the Left, the Grinding Wheel is Thinner than the Space between the Teeth and Grinds Only One Side of One Tooth at a Time

profile and of tooth thickness and is recommended where high precision is demanded. However, it requires numerous light cuts, and is therefore time-consuming.

Another method can be used when the volume of production, in combination with the number of teeth, does not justify the multiplication of indexing operations, and also when the type of cut and the size of the pitch radius precludes the application of one-sided grinding pressure. With the second method, the wheel is dressed to the thickness of the space, as shown at the right in Fig. 1, and grinds both sides simultaneously. The tooth thickness is predetermined by the dresser, which is mounted on the machine table in fixed relation to the work axis.

With helical gears, difficulties may be incurred because the profile on the wheel is not identical to either the transverse or normal section. The same problems exist as in the case of thread grinding because of the peculiar geometrical relationship between the wheel profile and that of the work. However, it is possible to predetermine the wheel profile on the basis of an exact mathematical method which strictly conforms to the characteristics of generation along a helical path.

Although there is no great difficulty in providing the involute profile on spur gears, problems arise at the root when the pitch is extremely fine. These problems are connected with grinding wheel breakdown at the tip when the width of the tip is extremely narrow. The experience of this company shows that the tip width of the wheel, which corresponds to the space width at the root diameter of the finished gear, should not be less than 0.005 inch. If some thought is given to the grain size of the grinding wheel (approximately 0.0025 inch for wheels of 220 grit), it will be appreciated that a space width of 0.005 inch represents an extremely successful performance of the wheel.

This condition appears to limit the range of fine-pitch work that can be finished by this method to about 120 pitch for a 20-degree pressure angle, and 180 pitch for a 14½-degree pressure angle. It should be recognized that grinding of very fine-pitch gears requires careful selection of the grinding wheels, thoughtful application of the coolant, and extreme care in the wheel dressing operation.

### *Gear-Tooth Tolerances Maintained in Form-Grinding.*

Gear-tooth tolerances that can be maintained by form-grinding are given in the accompanying table for standard precision master gears. The table shows a total variation of the involute profile of 0.0002 inch, half of which is provided for measuring errors of the checking equipment. Greater run-out accuracy than the values shown in the table can be obtained through selection.

Tooth-thickness tolerances can be readily liberalized or reduced as the case requires. Tolerances for ground fine-pitch work-gears range from the values given in the table up to the limits with which the gears are acceptable. The spacing, which is determined by the index-plate; the profile which depends on the set-up; and the parallelism, which is controlled by mounting conditions, need not be appreciably altered from this tabulation, even if production runs are contemplated. Greater tooth thickness and run-out tolerances allow for variations in the fit on the arbor, grinding wheel wear, etc.

VINCO STANDARD TOLERANCES FOR EXTERNAL MASTER SPUR GEARS.

| Gear Element | Tolerances, Inch* | |
|---|---|---|
| Tooth Thickness | | |
| Up to and including 4 inches pitch diameter | ±0.0001 | |
| 4 inches up to and including 8 inches pitch diameter | ±0.0002 | |
| Involute Profile | ±0.0001 | |
| Parallelism of Teeth with Axis | | |
| Up to and including 2½ inches face width | 0.0002 | |
| Over 2½ inches face width | 0.0003 | |
| Spacing of Any Three Consecutive Teeth | | |
| Up to and including 4 inches pitch diameter | 0.0002 | |
| 4 inches up to and including 8 inches pitch diameter | 0.0003 | |
| Run-out, Measured Over Standard Gear Wires, Total Indicator Reading | 14½ Deg. Pressure Angle | 20 Deg. Pressure Angle |
| Up to and including 4 inches pitch diameter | 0.0004 | 0.0003 |
| 4 inches up to and including 8 inches pitch diameter | 0.0005 | 0.0004 |
| Bore Diameter | | |
| Up to and including 1.500 inches diameter | +0.0002 | |
| 1.500 inches up to and including 4.500 inches diameter | +0.0003 | |
| Outside Diameter | | |
| Up to and including 4 inches pitch diameter | +0.001 | |
| 4 inches up to and including 8 inches pitch diameter | +0.002 | |
| Locating Face Run-Out—Total Indicator Reading | 0.0001 per inch of diameter (0.0001 min.) | |

*These tolerances suit the majority of requirements, but may be modified when necessary.

*Generating Method of Grinding Gears.*

The second principal method of grinding gear teeth is based on the process of generation with a straight-sided, rack-shaped tool. The generating motion is accomplished by two different arrangements. The first arrangement is embodied in the Pratt & Whitney or Maag gear grinders, which generate one tooth in one cycle and require indexing as in the case of form-grinding. Since each reciprocation of the wheel finishes only a very short element of the tooth profile, this method is usually slower than form-grinding. Its main advantages lie in the ease of controlling the profile of the grinding wheel, and its adaptability to the generation of gears that require a definite undercut, as well as helical gears. Since profile control and undercut are not critical in the case of fine-pitch gears, most of which are spur gears, this type of generation is not widely used for fine-pitch gears.

The second arrangement of generation by grinding is based on the principle of hobbing a threaded grinding wheel being employed in the same manner as a hob. It has the advantage of eliminating the indexing cycles, and is equally suitable for gears with and without undercut and for spur, helical, and spiral gears. The conception of this process is by no means new, the idea of using a threaded wheel dating back to 1890.

In this continuous generating arrangement, a timed relation is maintained between the rate of revolutions of the grinding wheel and the workpiece. The threaded grinding wheel has an axial section equal to the basic rack of the gear-tooth system. The linear pitch of the tooth spiral on the grinding wheel is equal to the circular pitch of the gear to be cut. The truing and dressing operation on the grinding wheel constitutes the most critical phase of this undertaking, and much thought and ingenuity have been invested in this phase of operations.

In this company's original studies of this arrangement, the fact was recognized that contact between the wheel and the work would be a spot contact, which travels along a spiral on the threaded wheel, if this process is used for finish-grinding, and that the perimeter of the wheel would be subjected to the greatest wear if it is used for gear-cutting. In order to distribute the contact with the

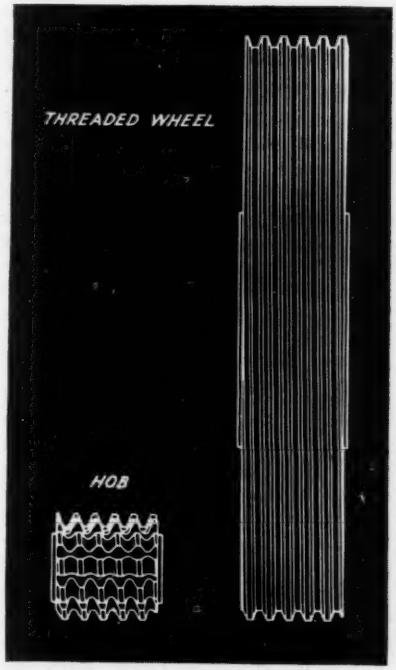



**Fig. 2. Threaded Grinding Wheel, Seen at the Right, Cuts Gear Teeth in the Same Manner as a Hob**

workpiece over the whole working area of the wheel, provision was made for either manual or mechanical axial adjustment of the wheel position, a feature that is analogous to the shifting of hobs. When this wheel adjustment is performed by a continuous mechanical motion, it is necessary, of course, to provide for a corresponding acceleration or deceleration of the work-piece.

*Preventing Wheel-Tip Breakdown.*

The risk of wheel-tip breakdown, which has a fundamental relation to the economy of the process as a whole, was minimized by the use of a grinding wheel of about 18 inches diameter. The large grinding wheel diameter has the desirable effect of reducing the helix angle on the grinding wheel, which, for 20 diametral pitch, would be only about 10 minutes. Errors caused by deviations of the grinding-wheel surface from a true involute helicoid are therefore negligible.

Notwithstanding the large diameter of the wheel, some measure of tip breakdown is to be expected. This was found to cause a rounding of the leading edge of the wheel-profile crest rather than a rapid outside diameter reduction. This peculiar tip breakdown leads to a lack of symmetry in the root shape of the workpiece which, if objectionable, can be controlled by reversing the direction of machine rotation.

A limitation of the use of the threaded wheel for generating gear teeth was found in connection with the diametral pitch. Up to this time, we have not been able to provide wheels for pitches finer than 64 because of space restrictions at the root of the wheel, which, for finer pitches, becomes inaccessible. A difficult problem is encountered even in the case of 64 pitch, if it is associated with a 20-degree pressure angle. Fewer difficulties, of course, are experienced when the pressure angle is 14½ degrees.

It should be possible to meet this problem by increasing the lead of the wheel so that it corresponds to a multiple of the circular pitch to be ground; in other words, by straddling two teeth in the work. This expedient can be resorted to, provided the number of teeth on the work is not a multiple of the number of pitches between wheel crests. The width of the crest would then present a secondary limitation with respect to very fine pitches.

There is, in principle, no limit with respect to the coarseness of the pitch, but it was found that this process, just as well as any other process of gear-cutting or gear generation, is more successful and requires relatively less effort when pitches are fine.

Our experience with this method of grinding covers a great variety of gears and gear materials. Fine-pitch gears down to about 20 pitch and perhaps finer are usually cut from the solid blank without additional finishing operations.

In the case of coarse-pitch gears, the process has been used for roughing and finishing purposes, and also as a salvage operation on heat-treated parts which were defective with regard to profile, spacing, or run-out. Adequate improvement of such defective parts was possible by free wheeling, without the use of the timing mechanism, in order to eliminate the problem of picking up existing tooth spaces.

Tolerances maintained by the use of this method are relatively close. They range from 0.0001 to 0.0002 inch on spacing; 0.0002 to 0.0003 inch on profile; and 0.0003 to 0.0005 inch on run-out. Statistical performance records are not yet extensive enough to establish definite tolerance charts and production time ratings.

It may be stated, however, that on the average, the use of this process has resuled in grinding time reductions up to about 80 per cent. Including set-up and dressing time, a cost reduction of about 30 to 50 per cent has been realized. In the case of fine pitches, especially where light metals or soft materials are cut, production time is comparable, although not superior, to that of hobbing. Generation grinding with a threaded wheel should be expected, therefore, to maintain its place, not only where hardened steel is involved, but also where better accuracy and finish may be desired without excessive additional cost.

# FOULING OF MARINE-TYPE HEAT EXCHANGERS.

BY H. E. BETHON, WASHINGTON, D. C.

ACKNOWLEDGMENT.

Presented on November 29, 1948 before the Annual Meeting of the American Society of Mechanical Engineers at the Hotel Pennsylvania, New York City, by Mr. Henry E. Bethon. Mr. Bethon is Assistant Head of the Heat Transfer and Industrial Gases Section of the Bureau of Ships, Navy Department.

## INTRODUCTION.

In order that the maximum striking or defensive power of a task force can be realized during a naval assignment, it is important that the design, construction, operation, and maintenance of major machinery components aboard naval vessels be standardized to the greatest extent possible so that each machinery component can be assured of operating at its peak efficiency. In other words, if there were no high degree of uniformity then certain machinery components, having different fouling characteristics, would be forced to shut down for cleaning or operate at reduced power and/or lowered efficiency. Toward this end the practice of the Bureau of Ships is to submit new designs of heat exchangers proposed by industry to governmental laboratories for the purpose of conducting performance tests in addition to evaluating fouling, cleaning, repair, and maintenance characteristics under conditions simulating actual shipboard operation before final approval is granted.

Heat-exchanger operation and maintenance aboard a naval vessel presents an ever-changing picture because, dependent on the vessel's operation in waters varying between the arctic and tropical regions, aside from operation in contaminated rivers and harbors, different marine growths and organisms and variations in chemical constituents are experienced, with the result that cleaning and operating methods must be varied to suit particular circumstances. Furthermore, naval vessels arriving in foreign ports of call often encounter different inferior grades of fuel and lubricating oils that adversely affect the performance of the related heat exchangers.

As regards heat-exchanger design, it safely can be assumed that there is a relatively fair degree of uniformity among manufacturers in the establishment of clean-tube heat-transfer rates as the result of the efforts toward standardization on the part of the Heat Exchange Institute and Tubular Exchanger Manufacturers' Association, among others. These rates are usually determined from velocity versus heat-transfer-rate curves adjusted for viscosity and temperature plus other physical variations, e.g., baffle type and pitch, tube length, diameter, and material,

etc., in the case of simple installations such as steam condensers, feedwater heaters, fuel-oil heaters, etc. In the case of complex installations wherein the temperatures of both fluids being handled vary, such as liquid-to-liquid exchangers, gas-to-liquid exchangers, gas-to-gas exchangers, etc., it is customary to employ the resistance method wherein the over-all clean-tube heat-transfer rate is equal to the reciprocal of the sum of the film resistance on the shell side plus the film resistance on the tube side plus the metal resistance.

However, very little has been done to establish a standard method for determining fouling resistances normal to the operation of the heat exchanger, with the result that a manufacturer may offer a design which does not allow adequately for a reasonable degree of fouling, either through lack of knowledge as to the fouling characteristics of the fluids being handled or to a strong endeavor to secure the contract. The result is that the purchaser finds it necessary to resort to frequent shutdowns of the plant for cleaning operations or to continue operation at reduced capacity and/or lowered efficiency. The initial design of a heat exchanger to meet the rated or designed capacity under the specified temperature and pressure conditions is important, but the ultimate successful operation of the unit is dependent primarily on the factors allowed for fouling and cleaning.

The heat exchanger should be designed so that the rated capacity will be realized after a specified number of hours under normal operating conditions has elapsed, thereby directly reflecting the period of satisfactory performance before cleaning becomes necessary. Under all circumstances the heat-exchanger manufacturer should indicate both the clean-tube and over-all heat-transfer rates. The Bureau of Ships has for years adopted this procedure in connection with distilling plants whereby the rated capacity of the plant is understood to be the output of the plant after 30 days of continuous operation, with clean-tube capacity arbitrarily established at 30 per cent in excess of rated capacity. Fouling-resistance rates are a function of the characteristics of the fluids being handled, velocity, tube material, temperatures, tube size, viscosity, among other items. Other factors besides fouling due to normal operation that must be considered in arriving at the optimum heat-exchanger design are weight, space, manpower, time outages, load factor, etc. For example, in the case of a submarine, where weight and space are at a premium, it is desirable to design a heat exchanger with reduced allowance for fouling, with consequent frequent cleaning periods, in order to keep weight and space at a minimum, whereas in the case of larger vessels where weight and space are not as critical, a greater allowance can be made for fouling, with the result that prolonged operation at rated capacity can be expected before cleaning becomes necessary.

*Design Considerations.*

In order to establish the proper design of heat exchanger which will be capable of sustaining the required capacity for the desired period of operation between cleaning periods with minimum maintenance and repair, the designer must give serious consideration to the selection of materials which offer maximum resistance to corrosion, to the fouling characteristics of the fluids being handled, and to the provision of adequate means for either manual or chemical cleaning of the unit when found necessary. In the case of heat exchangers, whose performance is greatly affected by fouling, the purchaser should be given as much information as possible on the proper chemical and/or mechanical methods to be employed in the cleaning of these units, in order that their maximum useful service life may be realized. By the same token the purchaser should acquaint the manufacturer fully with information as to the approximate period of storage under atmospheric conditions before installation, in order that the proper allowance for atmospheric corrosion can be made; as to conditions existing during installation that may result in unsuspected internal fouling of the unit from an outside source which, if serious

enough, may justify the adoption of a removable tube-bundle design or special cleaning features; as to possible abnormal sources of fouling or contamination not expected during normal operation of the unit which may be present, such as, seaweed, kelp, fish, barnacles, and the like. Other circumstances, not ordinarily apparent to the the manufacturer, which may have a strong bearing in determining the proper design of heat exchanger should be noted, particularly those items which will result in prolonged satisfactory service life with minimum maintenance, repair, and overhaul.

### *Causes of Fouling.*

Fouling of a heat exchanger may occur from the following causes among others:

(*a*) Fouling of transfer surfaces resulting from normal operation, such as deposition of scale, accumulation of slime, sludge, carbon deposits, etc.

(*b*) Fouling of transfer surfaces resulting from internal corrosion of ferrous subcomponents such as baffles, tubes, shell interiors, etc.

(*c*) Fouling of transfer surfaces resulting from foreign matter being carried into the heat exchanger by the circulating sea water, fuel oil, etc., from carelessness during the installation period, and from inadequate methods of cleaning the system before operation is started.

### *Design Factors That Minimize Fouling Problem.*

Aside from the importance of designing a heat exchanger with proper attention being paid to adequate allowance for fouling, selection of suitable materials, and to effective cleaning methods, every consideration should be given to establishing basic design requirements which will minimize the fouling or scaling problems. The following examples are cited:

(*a*) Design of distilling plants on the basis of vacuum operation wherein the last-effect shell vacuum is specified at 26.5 in. of mercury and the desuperheated steam to the first effect is maintained at subatmospheric pressures through the use of an orifice in the steam line. In this manner the temperatures throughout the system are held at low levels, thereby reducing scale formation.

(*b*) Adoption of satisfactory feed-treatment systems, such as the introduction of boiler compound and cornstarch, with the result that the formation of scale is greatly retarded, and the nature of the scale deposit is radically changed from a type which adheres rigidly to the tube surface and defies removal, to a type which is soft and powdery whose removal can be easily accomplished by means of a hosing-down operation. Current investigations are being undertaken on other methods of evaporator feed treatment such as ion exchange of the self-regenerating type, carbon-dioxide removal, electrolytic-cell installations, introduction of different types of chemicals, etc. It must be determined carefully beforehand that these feed-treatment systems will not result in accelerated corrosion or deterioration of the heat exchanger, or adversely influence the successful operation of the plant.

(*c*) Installation of a fresh-water-cooled lubricating-oil system for Diesel engines whereby the engine-jacket water, which has been treated by the introduction of an inhibitor, is employed to cool the engine lubricating oil, while the jacket water in turn is cooled by sea water. This arrangement requires only one sea-water-operated heat exchanger, thereby reducing the fouling and maintenance charges.

(*d*) Introduction of chlorine or chlorine compounds, the provision of a chlorine generating cell in sea-water piping systems or resorting to the periodic flushing of piping systems, including heat exchangers with heated sea water will reduce the contamination from marine growths and organisms.

(*e*) The use of low-viscosity high-grade fuel oils will reduce the operating temperatures necessary for proper atomization, with the result that less fouling of fuel-oil heaters is encountered.

This paper deals with fouling problems in connection with the following types of heat exchangers installed aboard naval vessels: Lubricating-oil and jacket-water coolers, steam condensers, fuel-oil heaters, and distilling plants.

## Lubricating-Oil and Jacket-Water Coolers.

### *Construction of Coolers.*

All main- and auxiliary-turbine lubricating-oil coolers installed aboard surface vessels are of the shell-and-tube type, better known as type A, with the lubricating oil circulated through the shell, and with the sea water circulated through the tubes. The main-turbine coolers, varying in size between 300 and 1600 sq. ft. are of the fixed tube-sheet design, as illustrated in Fig. 1 and are

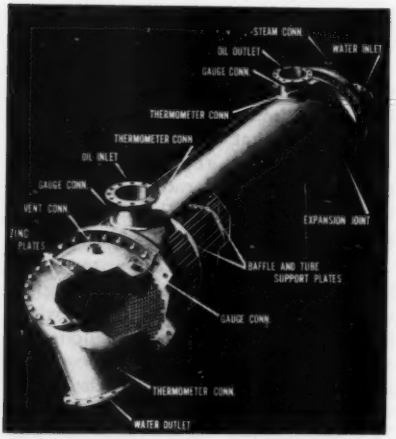



FIGURE 1.—Main-Turbine Lubricating-Oil Cooler, Alco Products.

constructed of ⅝-in.-OD 70:30 copper-nickel tubes, and 70:30 copper-nickel tube sheets, with cast bronze water boxes and with steel shell and baffles, wherein the baffles are either welded or bolted to the interior of the shell. Owing to difficulties experienced with the corrosion of the steel shell and baffles and, since it was impossible to inspect and/or clean these parts without retubing the unit completely, the Bureau of Ships, in 1944, required that the shell and baffles be made of nonferrous material and that the baffles be supported by tie rods and spacers attached to one tube sheet. Thus the entire baffle assembly could be withdrawn after the tubes were removed, for the purpose of cleaning the shell interior. Recently, for new construction, the specifications for main-turbine oil coolers have been changed to require a withdrawable tube-bundle design, based upon an outside packed floating tube sheet provided with telltale openings so as to detect immediately either a lubricating-oil or a sea-water leak. The auxiliary-turbine coolers, being much smaller in size, are either of the fixed or floating tube-sheet design and are usually constructed of ⅜-in.-OD 70:30 copper-nickel tubes and with non-ferrous shell and baffles.

The main and auxiliary Diesel-engine lubricating-oil and jacket-water coolers installed aboard all submarines and aboard the majority of Diesel-driven surface craft, one notable exception being LST's (landing ship, tank), are known as type B units with the lubricating oil or jacket water circulated through the tubes, and with the sea water circulated through the shell. These units are manufactured by the Harrison Radiator Division of General Motors Corporation and are furnished in many different sizes and models either of two basic groups, the largest unit being equivalent to a shell-and-tube unit containing approximately 400 sq. ft. of cooling surface. The two basic groups are plate-tube type and strut-tube type. Fig. 2 illustrates the

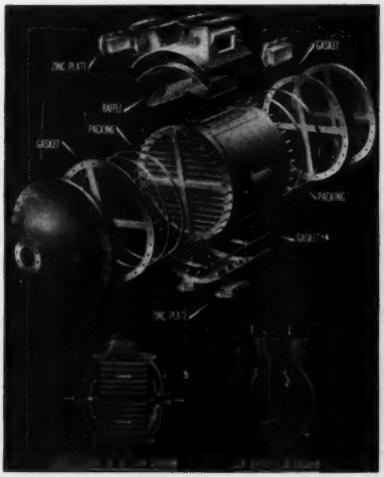



FIGURE 2.—Harrison Lube-Oil and Jacket-Water Coolers.

general principles of construction of a Harrision strut-tube-type cooler for an emergency Diesel-engine generator set.

The plate-tube cooler consists of a stack of plate-type tubes connected in parallel with the oil supply and enclosed in a cast-bronze housing through which sea water is circulated. In this design the plate assembly is removable. The fabricated strut-tube-type cooler comprises two different designs of strut tubes, one for water cooling and the other for oil cooling. The water tube is from 2 to 3 in. wide and is composed of a male and female section, each containing a series of formed dimples, which are copper-brazed together in a hydrogen furnace. The oil tube is 1 in. wide and is fabricated in the same manner as the plate-type tube, wherein the male and female sections, which enclose an internal distributor strip, are copper-brazed together in a hydrogen furnace.

The material of the plate- and strut-type tubes is 70:30 copper-nickel. The plate-type tubes are copper-brazed to a

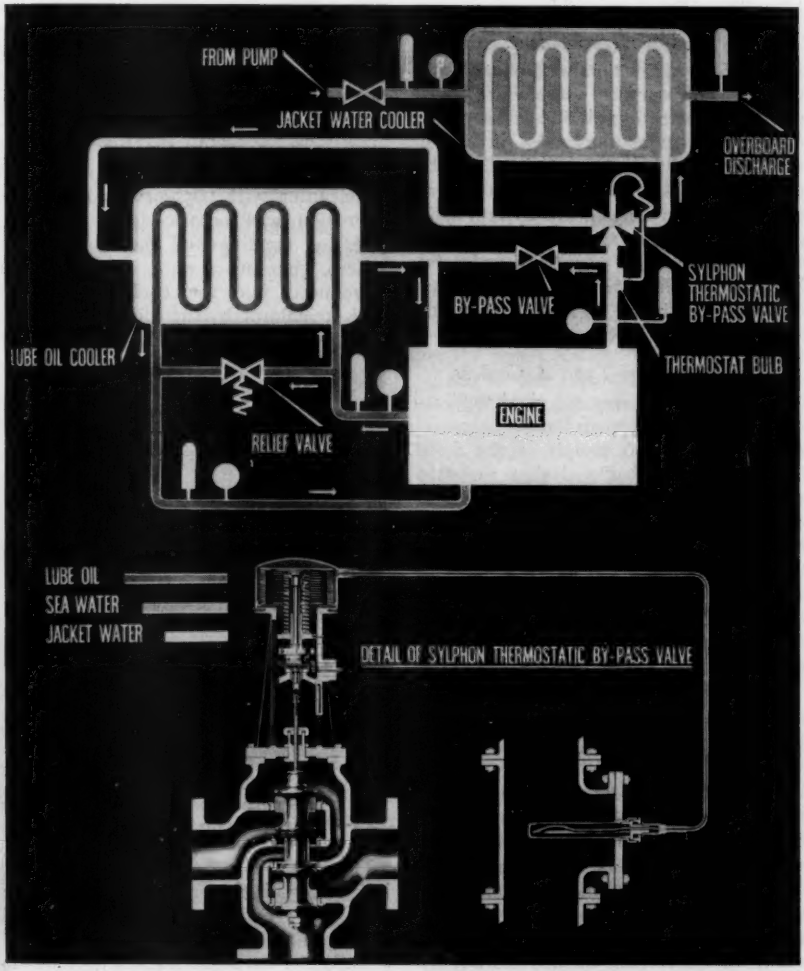



FIGURE 3.—Lube-Oil and Jacket-Water Cooling System.

70:30 copper-nickel header plate as an entire assembly. The strut-type tubes are soldered into a 70:30 copper-nickel header plate which in turn is soldered to the frame of the cast-bronze casing. All strut-tube-type models except Model HE24 are of the fixed tube bundle design. In the case of new construction, the shell or casing and oil or jacket-water heads or covers will be of brazed copper-nickel plate construction in lieu of castings, whereas the strut-type tubes will be silver-brazed in lieu of soldered to the header plates, and the tube-bundle assembly will be of the removable type.

*Piping Arrangement.*

Diesel-engine installations aboard submarines and surface vessels are based upon the lubricating-oil and jacket-water cooling system as illustrated in Fig 3, wherein the lubricating oil is cooled by the engine-jacket water which is in turn cooled by sea water. In this system sufficient by-passes and thermostatic controls are provided for the purpose of maintaining optimum lubricating-oil and jacket-water temperatures, regardless of engine loads and injection temperatures.

*Performance of Type A Coolers.*

The following cases are cited to illustrate the seriousness of fouled main-turbine lubricating-oil coolers of the fixed tube sheet and baffle design installed aboard various types of naval vessels which occasioned a large amount of additional work that would not have been necessary had these coolers been of a design which permitted access to clean them without retubing, or had they been constructed of noncorrosive material throughout.

U.S.S. *Shaw* (*DD373*). This vessel was damaged during the Pearl Harbor attack on Dec. 7, 1941, and the engine room flooded. The vessel was later run aground. Among other spaces, the engine room was flooded for the second time. Piping was thoroughly cleaned and lubricating-oil coolers were sent to the shop, cleaned with trichloroethylene solution, and tested afterward only hydrostatically. On completion of machinery work, the system was flushed thoroughly with oil several times, and the bearings were inspected and found satisfactory. After 6-hr. full-power operation at sea, the lubricating-oil pumps failed on the port unit.

On arrival at the yard with the port unit secured, the bearings in the port engine were inspected and found filled with abrasive rust, and many journals scored and bearings wiped. All upper gears were removed and sent to the shop for regrinding and resizing to common dimension; all bearings were remetaled. The lubricating-oil coolers were taken to the shop with instructions to open up and retube. When the tubes were cut out and coolers opened, excessive corrosion was found inside the wrapper sheet and on the baffles. The coolers were pickled thoroughly and baffles replaced.

This casualty was caused by the corrosion existing inside the lubricating-oil coolers, which had not been eliminated by cleaning with trichloroethylene. This corrosion was occasioned by the two previous floodings of the engine-room spaces, wherein sea water may have been introduced into the coolers. Cleaning with trichloroethylene did not remove this old rust. Fig. 4 shows the exterior

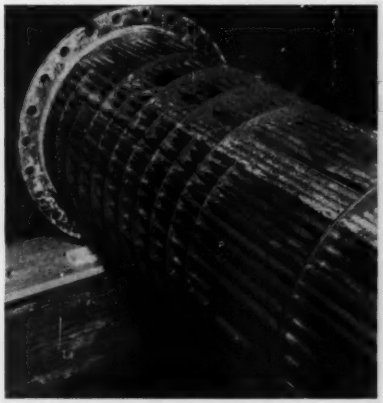


FIGURE 4.—Tube Bundle of Lube-Oil Cooler —Destroyer U.S.S. *Shaw* (DD373).

of the aft end tube bundle of the port lubricating-oil cooler. The baffles in the shell were badly corroded and coated with iron-oxide grit.

U.S.S. *Pensacola* (*CA24*). Due to damage and submersion in sea water, all

machinery units, piping, and lubricating-oil coolers in the after engine room were removed for cleaning and repairs. The two lubricating-oil coolers were completely disassembled, shells cleaned, tube sheets cleaned, and then the coolers were retubed. The coolers were reinstalled on board ship in March, 1943. About September 1, 1943, an examination was made of the oil and water sides of all coolers. Coolers in the forward engine room were found to contain much sludge and grit in the oil side of the shell. Coolers in the after engine room were found to contain rust and fine grit in sufficient quantity to warrant recleaning after having been installed about 5 months.

All coolers were taken to the shop, tubes removed, shells cleaned, reassembled, and retubed. After retubing, all coolers were tested to 150 psi with oil. All oil piping in both engine rooms was taken to the shop, cleaned, annealed, and reinstalled, followed by an extensive flushing schedule. Fig 5 shows the exterior of the tube bundle of the cooler in the after engine room. The baffles were badly corroded and heavily coated with iron-oxide grit.

FIGURE 5.—Tube Bundle of Lube-Oil Cooler —Cruiser U.S.S. *Pensacola* (CA24).

Numerous similar instances can be cited wherein main-turbine oil coolers were fouled to the extent of rendering them inoperative. Internal corrosion of the steel shell and baffles was caused by atmospheric conditions during prolonged periods in storage prior to installation aboard ship. Fouling was also caused by the lodgment of foreign matter such as rust, rags, weld beads, excessive dirt, chip drillings, etc., in the shell of the oil cooler, resulting from careless installation procedures. In some extreme cases the cooler tubes had to be removed completely in order to provide access to the shell side for cleaning purposes.

*Performance of Type B Coolers.*

In the case of type B (Harrison) lubricating-oil and jacket-water coolers wherein the flow passages are exceedingly small (less than ⅛ in.), strainers having a fine-mesh screen are usually provided in the sea-water circuits ahead of the cooler. No unusual fouling difficulties, except for rust scale, have been experienced in the lubricating-oil and jacket-water sides of these coolers during normal operation inasmuch as any foreign matter, the entry of which occurred during the ship construction period, will be deposited at the entrance to the tubes owing to the small flow passages, the removal of which can be readily accomplished.

In May, 1947, three Model 27 Harrison lubricating-oil coolers serving one main-turbine unit of the cruiser U.S.S. *Rochester* (CA124) completed over 4300 operating hours of satisfactory service performance without any leakage or fouling difficulties being encountered throughout the entire period. The three units were arranged in parallel on the oil side and in series on the sea-water side. No strainers were installed in the cooling-water system, and these units were not given any more operating attention than the standard shell and tube unit serving the adjacent main turbine. The great savings in weight and space of the Harrison units over the standard ⅝-in.-OD tube unit was particularly noticeable.

Just recently, the Harrison units were removed from the U.S.S. *Rochester* (CA124) and in their place has been installed one Ross ¼-in.-OD tube cooler, provided with a removable tube bundle, for the purpose of observing the fouling

characteristics of this unit under actual operating conditions. This cooler, along with a ⅜-in.-OD tube cooler of the same design and construction, recently passed performance tests at the Engineering Experiment Station, Annapolis, Md.

*Fouling of Coolers.*

The fouling of heat exchangers aboard naval vessels depends to a large extent on the sea-water characteristics in the region of the vessel's operations. Reports of considerable fouling of Harrison type and of small shell-and-tube-type heat exchangers serving auxiliary-machinery units have been received from the destroyer tenders U.S.S. *Dixie* (AD14) and U.S.S. *Piedmont* (AD17). Approximately 300 hr. of operation were obtainable between cleanings.

The warm and highly polluted waters of the operating area contributed to the rapid growth of fouling agents, which consist primarily of hydroids, shell fish (mussels), algae, and a mossy substance similar to fine grass roots. In both cases, strainers of sufficiently fine mesh are installed ahead of the heat exchangers. Fouling apparently is caused by embryo shell fish passing through the strainers and adhering to the interiors of the heat exchangers. Within 12 days after entry, the embryo fish grow to such proportions as to cause complete disruption of the flow of cooling water, with the resultant effect that in some cases machinery units have been rendered inoperative until cleared of the fouled condition.

The cleaning methods employed in the cases cited are strictly mechanized and manual. Low-pressure steam was employed to dislodge the growths and clean the surfaces down to bare metal. A diver was also sent overside periodically to remove the mass of marine growth from the sea strainers. The fouling is not excessive when there is a reasonably rapid flow of water through the heat exchanger.

In general, submarines have reported practically no fouling and very little scale on the sea-water sides of heat exchangers even after long hours in use; several instances have exceeded 5000 hr. with the result that these units yielded their full capacity output until they failed in service. It is believed that chemical action during acid-cleaning performed at a naval shipyard was responsible for their failure.

The submarine tender U.S.S. *Orion* (AS18) reported that the submarines under her care operated as much as 5000 hr. without cleaning and, except in the case of one ship which operated in the Mississippi River, no unusual operating conditions were encountered. Operations in the river caused mud to be deposited on the cooling surfaces, sufficient in some cases to cause complete stoppage. Main motors and main engines overheated at normal cruising speed on two engines. The major part of the mud fouling was cleared by submerging to 300 ft. and opening the cooler drains and later by blowing through with air at 225 psi.

The submarine tender U.S.S. *Nereus* (AS17) reported that upon removal of the eight main-engine jacket-water coolers and of the eight main-engine lubricating-oil coolers, both of which are of the shell-and-tube design (⅜-in.-OD), for cleaning and testing after the main engines operated an average of 2055 hr. each, approximately 20 per cent of the tubes were found to be plugged, and the interior of all tubes was coated with a white, hard scale averaging about 1/64 in. in thickness. The zincs were inspected and cleaned monthly and renewed when 50 per cent deteriorated. In addition, a considerable amount of marine growth and sediment was also present, despite the fact that strainers are installed in the sea-water lines ahead of all heat exchangers. Low and high sea-suction lines were found to be badly fouled throughout with marine growth. No ill effects or abnormal temperatures were noted due to the 20 per cent plugging of tubes or marine growth found in the heat exchangers. During the same overhaul period, the four auxiliary-engine jacket-water coolers and the four auxiliary-engine lubricating-oil coolers were removed for inspection. These units are of the Harrison design. The zincs were inspected and cleaned weekly and renewed when 50 per cent deteriorated. No unusual operating conditions existed other than the use of harbor water for pro-

tracted periods while at anchor. Upon inspections, these coolers showed small amounts of marine growth and sediment; no scaling was in evidence. There have been no ill effects or abnormal temperatures experienced due to scaling or marine growth.

*Operation of Coolers.*

Particular care must be taken in the operation of lubricating-oil coolers to avoid chilling the lubricating oil, which may result from full flow of cooling water and from by-passing the lubricating oil, in order to eliminate sludge precipitation which reduces the heat-transfer rate, thus necessitating frequent cleanings. Furthermore, too low an oil temperature (below 130 deg. F.) may result in getting condensation into the bearing housings and gear casings. This entry of water into the lubricating-oil system sooner or later will prove troublesome. The cruiser U.S.S. *Topeka* (CL67) reported the failure of worm-gear casings on the condensate and booster pump units due to operating with too cold an oil temperature.

In order to minimize the fouling and corrosion of the sea-water side of the Harrison type jacket-water coolers installed aboard the LSM's (landing ship, medium), it was found desirable to provide built-in air-separating tanks, connect the salt-water side of the main-engine fresh-water coolers in parallel, coat the interiors of the sand traps and air-separating tanks and all surfaces of the strainer screens in the sand traps with antifouling paint, and install water-actuated sand eductors in the sand traps among other corrective measures.

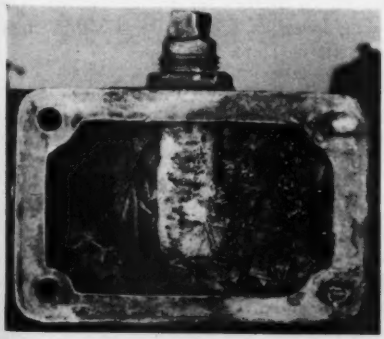


FIGURE 6.—Fouled Condition of Small Harrison Diesel-Engine Water Cooler.

Fig. 6 illustrates the extent to which a small marine Diesel-engine water cooler was plugged with lint and rags on the water side in the vicinity of the zinc pencil, which probably was caused by careless installation procedures.

FIGURE 7.—Fouled Condition on Water Side of Harrison Model HE424-362 Oil Cooler.

Fig. 7 illustrates the fouling of a Model HE424-362 Harrison strut-tube-type lubricating-oil cooler removed from the seaplane tender U.S.S. *Barnegat* (AVP10) after being in service over 3 years. The fouling on the inlet water side with stones, mud, and silt can be seen plainly.

Fig. 8 illustrates the fouling of a Model 60 Harrison plate-tube-type lubricating-oil cooler also removed from the seaplane tender U.S.S. *Barnegat* (AVP-10) after being in service over three years. The fouling of the plate tubes on the water side with marine growth can be observed readily.

Fig. 9 illustrates the fouling on the lubricating-oil side with carbon sludge of a Model HE124-250 Harrison strut-tube-type lubricating-oil cooler removed from the tug *Sheila Moran* after being in service approximately 15 months.

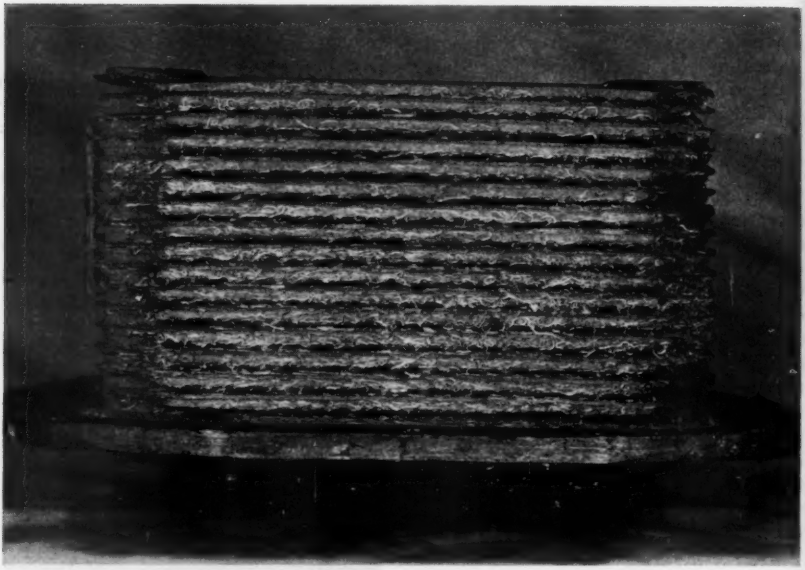


FIGURE 8.—Fouled Condition on Water Side of Harrison Model 60 Oil Cooler.

## STEAM CONDENSERS.

### *Construction of Condensers.*

The main condensers installed aboard combatant naval vessels and the majority of large turbine-driven auxiliaries are of single-pass construction employing 5/8-in.-OD copper-nickel tubes wherein the circulating water is provided by a scoop injection system. A circulating pump, taking suction from a separate sea chest, is provided for low speeds and for astern operation. The main condensers installed aboard vessels built to the specifications of the U. S. Maritime Commission which were later converted to naval auxiliaries are, in general, of two-pass construction employing 3/4-in.-OD aluminum-brass tubes, in which the circulating water is provided by a circulating pump. The main condensers vary in size from 3,800 sq. ft. to 28,000 sq. ft., the latter size being installed aboard the CVB41 class aircraft carriers.

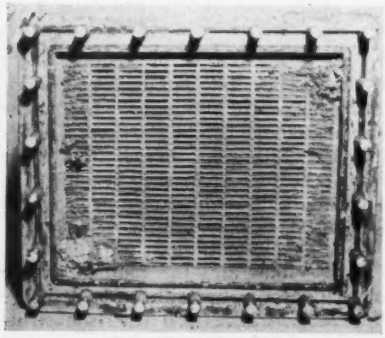


FIGURE 9.—Fouled Condition on Oil Side of Harrison Model HE124-250 Oil Cooler.

Fig. 10 illustrates the main condenser provided for the BB61 class battleships. The water boxes are of nickel-copper (monel) welded-plate construction of the sectional type, the tubes and tube sheets are of (70:30) copper-nickel material, whereas the shell-and-tube support plates are of steel plate. The main condensers are installed either athwartship or fore and aft dependent upon whichever location best suits the arrangement of machinery in the engine rooms. The dynamo or auxiliary condensers are of

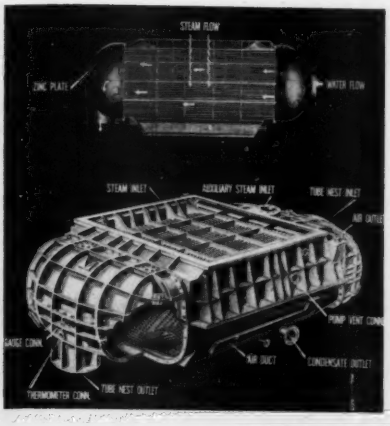



FIGURE 10.—Main Condenser—BB61 Class Battleships, Foster Wheeler.

two-pass construction employing the same materials as in the case of the main condensers, except that the water boxes are of cast bronze in lieu of monel metal. Fig. 11 illustrates the auxiliary condenser provided for the DD445 class destroyers.

*Fouling of Steam Side.*

Ordinarily the fouling of the steam side of the condenser under normal operating conditions presents no undue difficulties. The removal of any grease or dirt is periodically accomplished by boiling out with a strong solution of Navy standard boiler compound at intervals of every 2 or 3 years. In order to eliminate atmospheric corrosion of the shell interiors, the Bureau of Ships, in 1944, required that a coating of corn oil and Japan drier or an equivalent protective

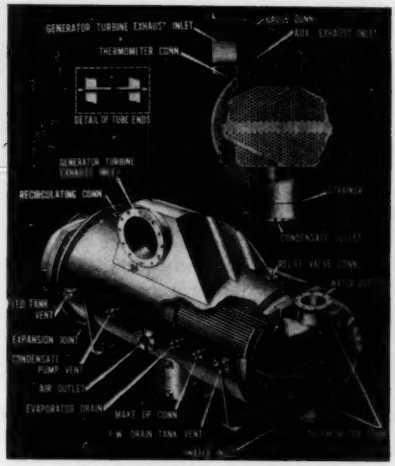



FIGURE 11.—Auxiliary Condenser—DD445 Class Destroyers, Foster Wheeler.

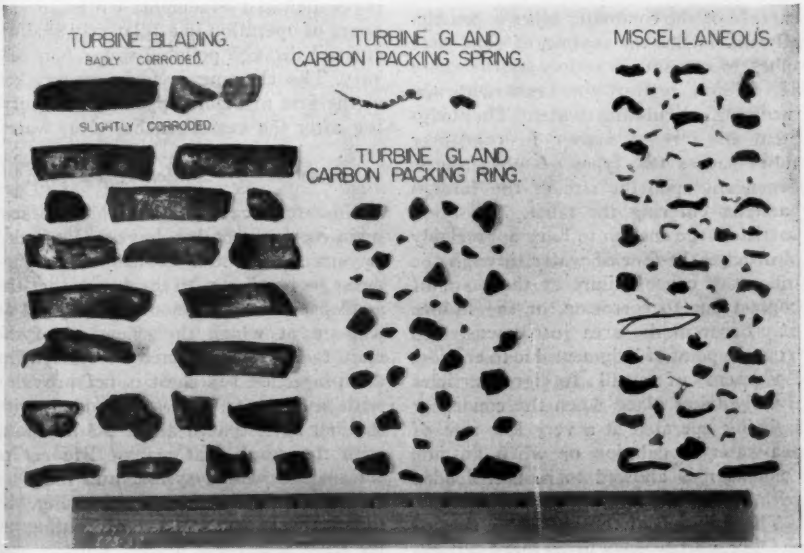



FIGURE 12.—Foreign Materials Removed From No. 1 Dynamo Condenser U.S.S. *Wichita* (CA45).

coating be applied to the inside of the steel-plate shell and to the steel tube-support plates. Under these circumstances it is possible to maintain condensers satisfactorily in open storage for extended periods without incurring any serious corrosion of the condenser shell. Fig. 12 illustrates the foreign materials removed from No. 1 dynamo condenser aboard the U.S.S. *Wichita* (CA45) resulting from a casualty to No. 1 turbo-generator. The foreign materials comprise primarily pieces of turbine blading and of carbon packing ring.

### *Fouling of Water Side.*

On the other hand, it is extremely important to maintain the inside of the condenser tubes free from fouling inasmuch as the service life of the condenser tubes is greatly dependent thereon.

Under normal operating conditions, the fouling is not severe enough to affect seriously the operating efficiency of the prime mover but usually results in accelerated tube failures which require the turbine to be secured for the purpose of plugging or replacing the leaky tubes. Owing to the low circulating-water temperatures and to the favorable scaling characteristics of copper-nickel tubing, the amount of scale formed on the inside surface of the condenser tubes is practically nil. Hence the fouling of condenser tubes is occasioned mainly by the entry of foreign matter entrained with the incoming circulating water. The lodgment of foreign matter in condenser tubes causes two types of tube failure, dependent upon the size of the foreign particles entering the tubes. A foreign particle large enough to have appreciably obstructed the flow of water through the tube may cause failure at the point of contact due to corrosion, or the failure may occur in the area just downstream from the point of lodgment due to erosion. Lodgment of small foreign particles usually takes place when the condenser is being operated at a very low rate of sea-water circulation or when an idle condenser is allowed to remain flooded with polluted harbor or river water.

The draining of the sea-water side of an idle condenser may prove more dangerous than leaving the condenser flooded with polluted water, unless care is exercised that the condenser is properly cleaned at the time of draining and the tubes left dry. In the event the condenser is drained and allowed to stand without cleaning, polluted water in low points along the tube will evaporate gradually, concentrating the impurities which are left behind as the evaporation proceeds. Thus highly polluted water, concentrated in small areas along the bottom of the tubes, corrodes and grooves the tube wall in these areas. When condensers are drained in port, it is highly desirable that the sea-water side be washed out thoroughly with fresh water. If ample fresh water is available, the most desirable method is to water-lance each tube, using fresh water.

During the summer of 1944, the Philadelphia Naval Shipyard was very much concerned with the serious water conditions in the Delaware River resulting from the generation of hydrogen sulphide fumes arising from the decomposition of untreated sewages and waste. As a result of this contaminated water condition, the Bureau of Ships issued separate instructions requiring the inspection and cleaning of condenser tubes in the case of vessels proceeding to sea from ports such as Philadelphia where several hours of operation in a relatively shallow channel through polluted water is necessary. The cleaning is to be undertaken at the first available opportunity occurring after the vessel reaches clear water.

### *Examples of Condenser Fouling.*

Numerous cases are on file wherein main-condenser tubes have been fouled by marine life such as fish, eels, "Japanese worms," etc., to the extent that the condenser vacuum suddenly dropped to a point at which the efficiency of the main turbine was affected adversely. The main-injection sea chest is not provided with a strainer but is provided with strainer bars, spaced about 2-3 in. apart, with the result that marine life enters the main injection system and fouls up the condenser-tube entrances. Fig. 13 illustrates a severe case of fouling at the inlet tube sheet of No. 1 main con-

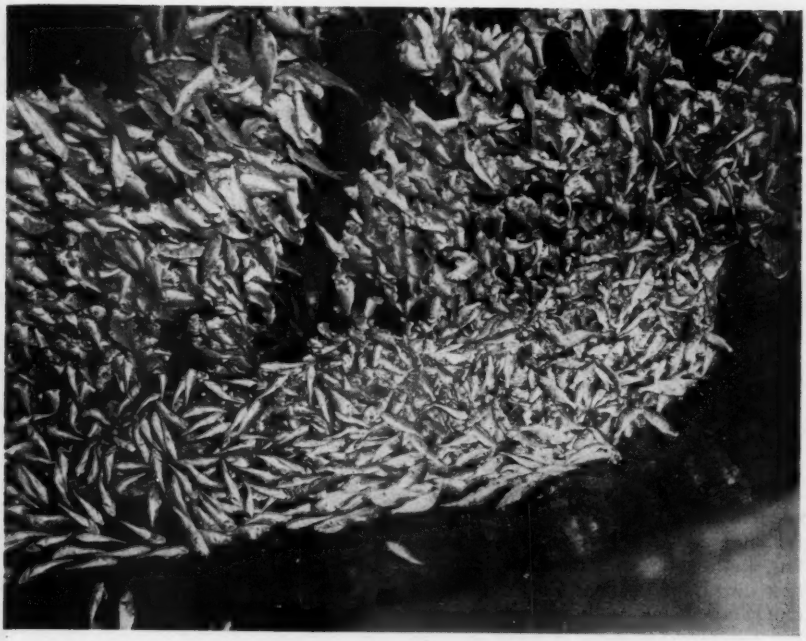


FIGURE 13.—Fish Trapped at Inlet Tube Sheet of Main Condenser—U.S.S. *Saipan* (CVL-48).

denser installed aboard the aircraft carrier U.S.S. *Saipan* (CVL48). This view was taken on December 3, 1947, and judging from the diameter of the belled ends of the plastic condenser-tube inserts, it can be seen readily that these fish are on the average about 10-12 in. in length.

The destroyer U.S.S. *Fullam* (DD-474), on June 17, 1945, reported that after completing 78 days' steady steaming in the Ryukus Area the No. 1 main condenser was drained and opened and thousands of living, white squirming "worms" were found sticking to the sides of the water box, tube sheets, and, most important, inside the tubes themselves. None of this marine growth was found in an attached condition on the discharge side. It is estimated that these organisms were found in 40 to 50 per cent of the tubes. Samples of this marine growth were forwarded to the Bureau of Ships and were chosen to show progressive stages of growth which varied from infinitesimal white spots to a growth approximately ⅜ in. diam. and 3 to 4 in. long. In the advanced stages of growth the parasite begins to resemble a snail; the living end beginning to harden and having whorls similar to a snail. The outstanding point of this growth is the adhesive pad of fibrous material by which the growth attaches itself to the condenser, necessitating that the growth be scraped or peeled from the condenser surfaces. Samples of this growth were later identified as a Japanese barnacle known technically as *Conchoderma Auritum*.

The condenser was first steamed to kill the growth, the surfaces were then cleaned carefully using a light scraper on the water box. A scrubbing brush and fresh water were used on the tube sheets, and then the tubes were steam-lanced, followed by air-lancing from the discharge side. The debris was cleaned, portable blowers were rigged, and the condenser was allowed to set over night. In the morning the air-lancing was repeated. With the completion of this process, investigation showed less than 5

per cent of the tubes had any obstruction remaining. Instructions were issued to effect the periodic overheating of the condenser circulating water to a temperature of about 5 deg. F. above the maximum growth-environment temperature, with the intent of killing the barnacle, thus promoting subsequent rotting and washing away of the remains while under way.

During wartime operations, unusual fouling of the main-injection circulating-water systems and of the distilling-plant-inlet sea chests with fuel oil and its resultant by-products were experienced, especially after a naval vessel passed through a heavy oil slick, resulting from the previous sinking of an oil tanker. In the case of one tanker, a tarry substance approximately ¼ in. thick covered the inlet tube sheet over large areas and consequently blocked the flow of circulating water through the condenser tubes. In the case of distilling plants, the evaporator tube nests became fouled with a heavy coating of oil, and the units had to be boiled out thoroughly before the resumption of normal operation

## Fuel-Oil Heaters.

### *Construction of Heaters.*

Since 1938, all naval vessels have been furnished with type B fuel-oil heaters wherein the heating steam passes through the inner tubes which contain external fins, and the fuel oil passes in a counter-flow direction along the fins between the outer tubes and the inner tubes. Since the resistance to heat transfer is preponderantly on the oil side, it is therefore economical to provide extended finned surface in order to keep the weight and space at a minimum. Fig. 14 illustrates a sectional G-fin fuel-oil heater manufactured by the Griscom-Russell Company. In this design the inner tube sections are removable for the purpose of replacement or mechanical cleaning. In general, the fuel-oil heating installation is divided between two or more sections operating in parallel so that each section performs at its maximum capacity under varying load demands, thereby reducing unnecessary fouling of the heaters. Before type B fuel-oil heaters are accepted by the Bureau of Ships, they must pass exhaustive tests at the Naval Boiler and Turbine Laboratory to determine their performance and fouling characteristics in addition to weight and space evaluations. Conventional shell-and-tube-type fuel-oil heaters, known as type A, are installed aboard naval vessels built prior to 1938, and aboard mechant vessels converted to naval auxiliaries.

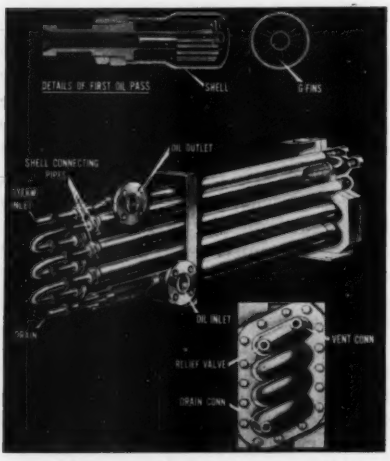



Figure 14. — Sectional G-Fin Fuel-Oil Heater, Griscom-Russell.

### *Fouling of Heaters.*

The rate of fouling on the oil sides of fuel-oil heaters is governed by three factors, namely, the rate at which oil passes through the heater, the temperature to which the oil is heated, and the nature of the oil itself. Overheating of the oil is probably the greatest single cause of fouling. "Cracked" oils, particularly, break down under heat and deposit carbon residue; such oils are more reduced in viscosity for a particular rise in temperature than straight-run oils. Under these circumstances, it is especially important that cracked oils should not be heated beyond the temperature which gives the correct viscosity for efficient atomization. Care must be taken when

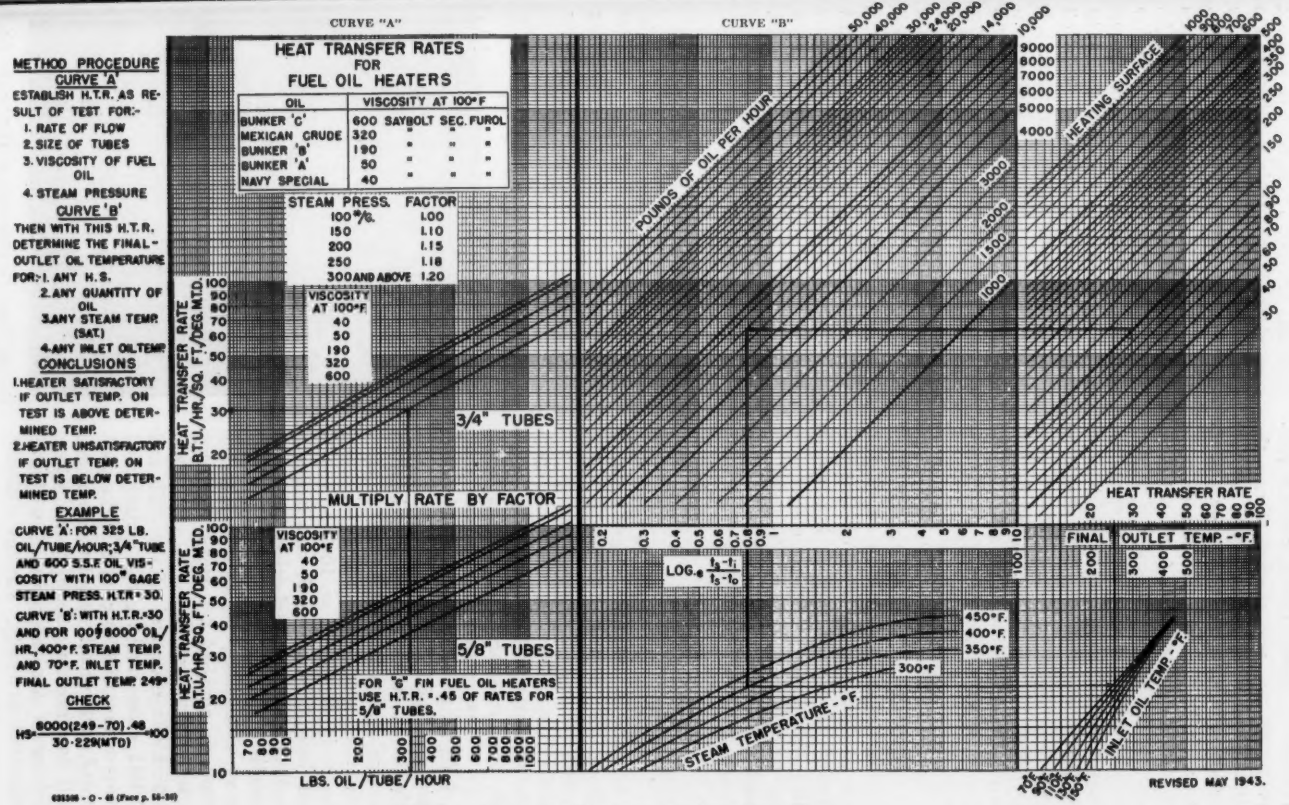

METHOD PROCEDURE
CURVE 'A'
ESTABLISH H.T.R. AS RESULT OF TEST FOR:-
1. RATE OF FLOW
2. SIZE OF TUBES
3. VISCOSITY OF FUEL OIL
4. STEAM PRESSURE
CURVE 'B'
THEN WITH THIS H.T.R. DETERMINE THE FINAL-OUTLET OIL TEMPERATURE FOR:- 1. ANY H.S.
2. ANY QUANTITY OF OIL
3. ANY STEAM TEMP. (SAT)
4. ANY INLET OIL TEMP.
CONCLUSIONS
1. HEATER SATISFACTORY IF OUTLET TEMP. ON TEST IS ABOVE DETERMINED TEMP.
2. HEATER UNSATISFACTORY IF OUTLET TEMP. ON TEST IS BELOW DETERMINED TEMP.
EXAMPLE
CURVE 'A': FOR 325 LB. OIL/TUBE/HOUR; 3/4" TUBE AND 600 S.S.F. OIL VISCOSITY WITH 100# GAGE STEAM PRESS. H.T.R = 30.
CURVE 'B': WITH H.T.R. = 30 AND FOR 10,000 LB. OIL/HR., 400° F. STEAM TEMP. AND 70° F. INLET TEMP. FINAL OUTLET TEMP. 249°
CHECK
HS = 8000(249 - 70) .48 / 30 - 22.5(MTD) = 100
HEAT TRANSFER RATES FOR FUEL OIL HEATERS
OIL | VISCOSITY AT 100° F
BUNKER 'C' | 600
MEXICAN CRUDE | 320
BUNKER 'B' | 190
BUNKER 'A' | 50
NAVY SPECIAL | 40
SAYBOLT SEC. FUROL
STEAM PRESS. | FACTOR
100 #/S. | 1.00
150 | 1.10
200 | 1.15
250 | 1.18
300 AND ABOVE | 1.20
VISCOSITY AT 100° F.
40
50
190
320
600
HEAT TRANSFER RATE B.T.U./HR./SQ. FT./DEG. M.T.D.
3/4" TUBES
MULTIPLY RATE BY FACTOR
5/8" TUBES
FOR "S" FIN FUEL OIL HEATERS USE H.T.R. × .45 OF RATES FOR 5/8" TUBES.
LBS. OIL/TUBE/HOUR
CURVE "A"
CURVE "B"
POUNDS OF OIL PER HOUR
HEATING SURFACE
HEAT TRANSFER RATE
FINAL OUTLET TEMP. - °F.
LOG e (t2 - ti) / (ts - to)
450° F.
400° F.
350° F.
300° F
STEAM TEMPERATURE - °F.
INLET OIL TEMP. - °F.
REVISED MAY 1943.

FIGURE 15.—Fuel-Oil-Heater Performance Chart.

heaters are being operated at reduced capacity to insure that excess oil temperatures are not reached. Since the normal rate of operation is low, the fact that heavy fouling exists may not become apparent before an attempt is made to operate the heater at its full rated capacity. To determine the state of fouling during operation at reduced rates, the curves shown in Fig. 15, which were devised by the author in 1935, should be used for predicting fuel-oil-heater performance at full power. When performance checks indicate that a heater has fouled, it should be cleaned at the first available opportunity.

Several destroyers which have just recently returned from operations in the Mediterranean reported a serious fouling condition in the case of Griscom-Russell G-fin type fuel-oil heaters. In each instance much lint had collected on the fins which were visible from the header. Also a thin carbonaceous deposit had collected on all surfaces in contact with the fuel oil. These heaters had not been cleaned since their installation 3 to 4 years ago. The cause of fouling can be attributed directly to the type of oil received in the Mediterranean area. This fuel oil contained much water and insoluble material, some of the latter showed up in the fuel-oil trunk screens in the form of large wads and had to be removed to continue fueling. This oil also had a high viscosity, with the result that it had to be heated to a higher temperature, approximately 180 deg. F. The heaters would not have been so fouled if a strainer had been installed ahead of the unit. The oil strainers located on the discharge side did not become excessively fouled with foreign matter.

Owing to the improved grades of fuel oil employed by the Navy, very little difficulty has been experienced with the fouling of the oil sides of fuel-oil heaters.

## Distilling Plants.

*Construction of Distilling Plants.*

In general, distilling plants installed aboard all steam-turbine-driven naval vessels since 1922, have been of the multi-effect, low-pressure submerged-tube type, designed to operate on auxiliary exhaust steam at pressures below 5 psi. These plants vary in size from 4, 8, 10, 12, and 20,000 gpd. double-effect units to 20, 30, and 40,000 gpd. triple-effect units. In the case of Diesel-driven surface vessels and submarines, motor-driven vapor-compression distilling plants, in capacities of 1000 and 2000 gpd. have been installed, except for a small number of Diesel-driven auxiliary vessels which are provided with low-pressure distilling plants in which the steam is supplied by small auxiliary steam-generating units.

With the exception of the CV9 and CVB41 class carriers, the majority of low-pressure distilling plants installed aboard naval vessels are of the combined shell construction whereby certain heat exchangers are incorporated within a single-unit assembly. Various different arrangements of distilling-plant combinations have been designed by Griscom-Russell Company, Foster Wheeler Corporation, New York Shipbuilding Corporation, and Bethlehem Steel Company.

Fig. 16 illustrates a double-effect "solo-

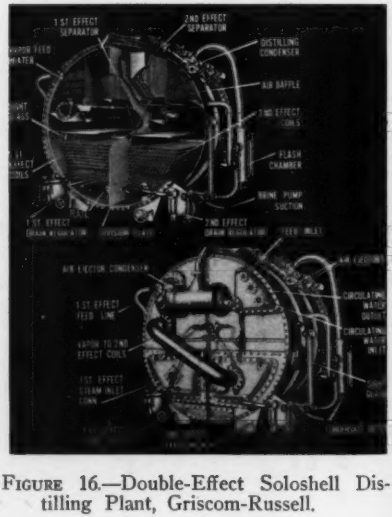



FIGURE 16.—Double-Effect Soloshell Distilling Plant, Griscom-Russell.

shell"-type distilling plant that is furnished in the 4, 8, 10, and 12,000 gpd. sizes. The only external unit not shown in this assembly is the condensate cooler. The first- and second-effect evaporators, vapor feed heater, distilling condenser, and vapor baffles and separators are all contained in a single shell with a vertical division plate separating the two effects. The air ejectors, air-ejector condenser, flash chamber, and drain regulators are supported externally by the shell.

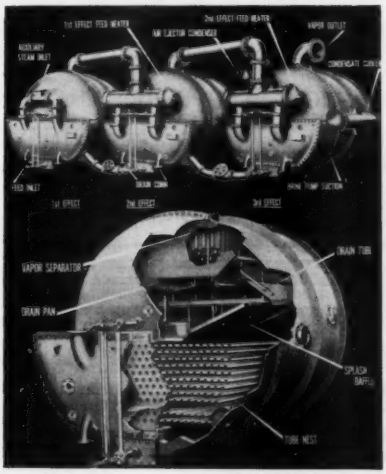


FIGURE 17. — Triple-Effect Separate Shell 40,000-Gpd Distilling Plant, Griscom-Russell.

Fig. 17 illustrates the 40,000 gpd. triple-effect separate-shell distilling plant provided on the CV9 and CVB41 class carriers. The evaporators (3), vapor feed heaters (2), distilling condenser, air-ejector condenser, and condensate cooler are all independent units which are separately piped up to complete the assembly of the distilling plant. The vapor baffles and separators are incorporated within the evaporator shell for each effect.

The material of the evaporator shells is cast bronze or naval rolled brass, except that in the more recent units, welded copper-nickel plate is employed. The material of the tubes and tube sheets in the condensate cooler, distilling condenser, air-ejector condenser, and vapor feed heater(s) is 70:30 copper-nickel. Admiralty tubes and naval brass tube sheets were employed in the construction of the evaporator tube bundles but the Bureau of Ships is now requiring 70:30 copper-nickel for these component parts.

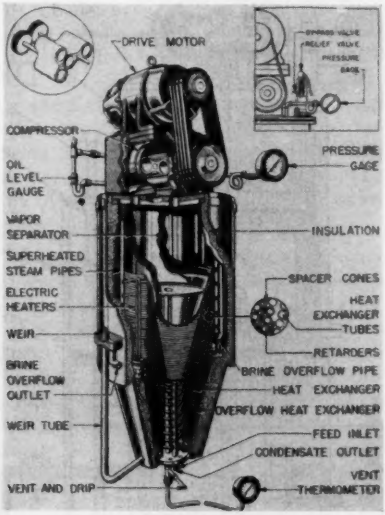
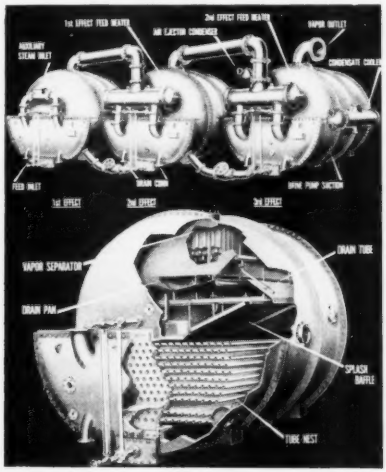



FIGURE 18.—Kleinschmidt Model S Vapor-Compression Distilling Unit

Fig. 18 illustrates the Kleinschmidt model "S" vapor compression distilling plant which was installed aboard submarines and Diesel-driven surface craft from 1940 to 1943, whereafter the Badger model "X" plant, as shown in Fig. 19, was substituted in its place. In the model S plant the evaporation-condensation and feed heating-condensate cooling are accomplished in the same unit. In the Model X plant these functions are handled by two independent units known as the evaporator and the heat exchanger. In the case of the model X plant straight-tube construction is used throughout, which permits mechanical cleaning, whereas, in the case of the model S plant chemical cleaning is favored, although the unit can be mechanically cleaned. However, considerable time is required to dismantle and clean the ten conical-shaped coil assemblies. The material employed throughout the construction of both model S and X plants is 70:30 copper-nickel.

shell"-type distilling plant that is furnished in the 4, 8, 10, and 12,000 gpd. sizes. The only external unit not shown in this assembly is the condensate cooler. The first- and second-effect evaporators, vapor feed heater, distilling condenser, and vapor baffles and separators are all contained in a single shell with a vertical division plate separating the two effects. The air ejectors, air-ejector condenser, flash chamber, and drain regulators are supported externally by the shell.

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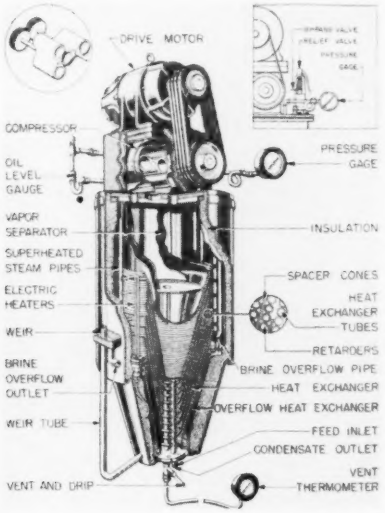


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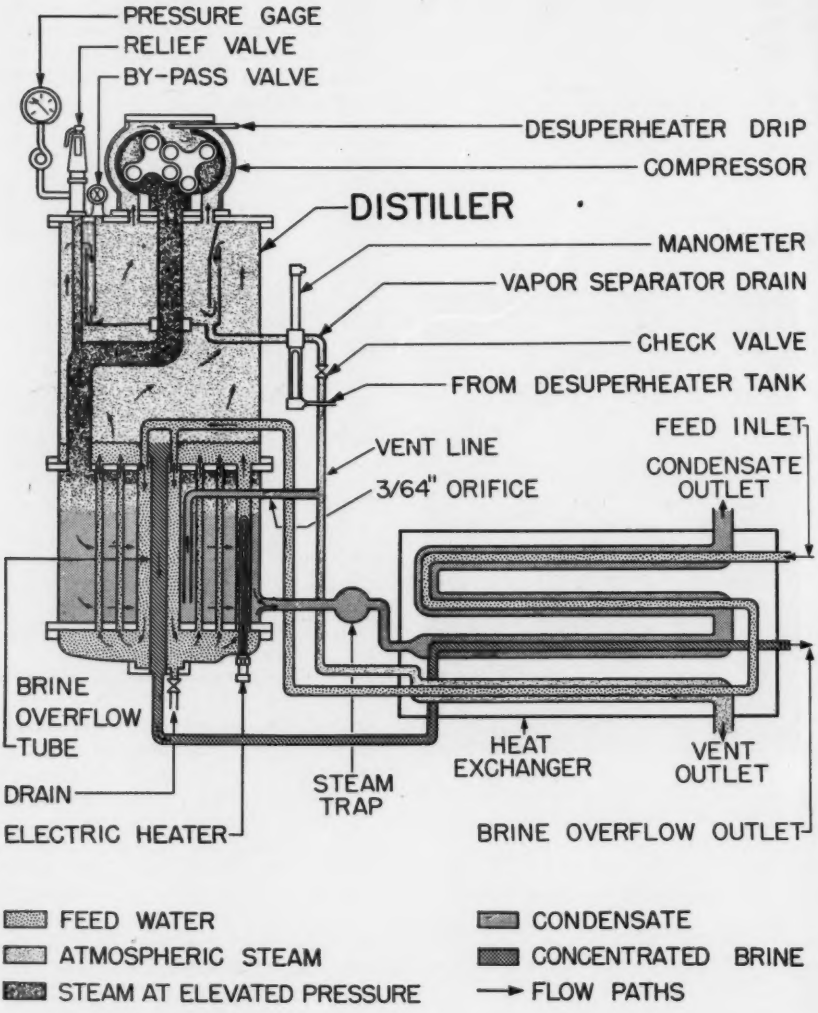



FIGURE 19.—Badger Model X-1 Vapor-Compression Distilling Unit.

*Fouling of Distilling Plants.*

The principal cause of fouling in distilling plants is the deposition of scale on the tube transfer surface of the evaporators and associated heat exchangers. The rate of scale formation is affected by the operating temperatures, brine density, and nature of the solids in the feed among other factors. In order to keep scale formation at a minimum, the design of the low-pressure steam distilling plant is based on producing the rated capacity with subatmospheric steam pressures (below 212 deg. F.), and on maintaining a brine density in the last effect shell not over 1.5 thirty-seconds. In addition

to these design requirements, the Bureau of Ships has required the installation of a continuous cornstarch-boiler compound feed-treatment system. It is recognized that one of the best ways to eliminate or reduce scale formation is through the process of evaporator feed treatment, and numerous research and development projects as hereinafter described have been established toward achieving this goal.

Rear Admiral Thorvald A. Solberg, now Chief of the Office of Naval Research, conceived and initiated the idea of treating the evaporator feed aboard United States naval vessels. During 1941, while he was Force Engineer of the Battle Forces, Pacific Fleet, actual experimental shipboard installations were made. Prior to this period, he conducted numerous laboratory tests on board ship in an attempt to arrive at the most practical method of treating evaporator feed that would obviate the need of carrying enormous supplies of chemicals aboard naval vessels. As a result, a mixture of boiler compound and cornstarch was adopted as the most satisfactory solution for low-pressure-steam distilling plants.

In a recent survey, the Boston Naval Shipyard reported that no unusual scaling was noted in vapor-compression or low-pressure-steam distilling plants. However, it was indicated that certain ships currently in the yard for overhaul have scale of appreciable thickness, about ¼ in. Other ships operating under similar circumstances and with like periods of cleaning and overhaul of their evaporators, namely, three years, have relatively clean plants. The evaporators on most of the ships mentioned have never been removed and cleaned since their original installation, 3 to 4 years ago. A physical inspection revealed that the scale ranged from 1/32 to 1/4 in. thick, the thicker layer being on the shell, which is usually the case. One ship reports that the evaporator capacity was not seriously impaired by this scale. All ships stated that they have followed the shipboard procedure of using cornstarch-boiler compound and cold-shocking the tubes for preventive maintenance.

Vessels operating in the Mediterranean area reported that due to the temperature and high carbonate content of the evaporator feed, the tubes become fouled in a very short time. One submarine, after returning from a 108-day patrol in the Pacific area, reported that the distilling plants were operating at near capacity after being in use for about 80 days of the patrol. Since adequate strainers are provided in the circulating-water supply lines of both law-pressure-steam and vapor-compression distilling plants, no unusual conditions of fouling due to entry of foreign matter or marine growth have been experienced.

An excellent argument against the use of ferrous material in the construction of distilling plants is indicated in Fig. 20, (photographed in May, 1938), which shows the condition of the interior of the steel-plate shell of the combined distilling-condenser unit installed on board the destroyer U.S.S. *Clark* (DD361). It should be mentioned that this extensive corrosion took place within less than 2 years after completion date of the vessel and occurred on the fresh-water side of the unit, since the sea water is circulated through the tubes. The condition of the combined distilling-condenser units installed on the other destroyers of this class was essentially the same, indicating that this condition was general. As a result of this experience, the Bureau of Ships authorized the replacement of all steel-plate distilling-condenser shells aboard the DD356-363 class destroyers and aboard the U.S.S. *Brooklyn* (CL40) class cruisers with shells of cast-bronze construction, and in addition required nonferrous construction for all future distilling-plant installations.

Figs. 21, 22, and 23 illustrate the progressive degree of tube fouling of the air-ejector after condenser, inner feed heater and vapor feed heater, respectively, comprising the No. 1 distilling plant installed on board the cruiser U.S.S. *Brooklyn* (CL40). These revealing views were taken in September, 1944, before the installation of the cornstarch-boiler compound evaporator-feed treatment system. The total months in operation since the last cleaning of the after

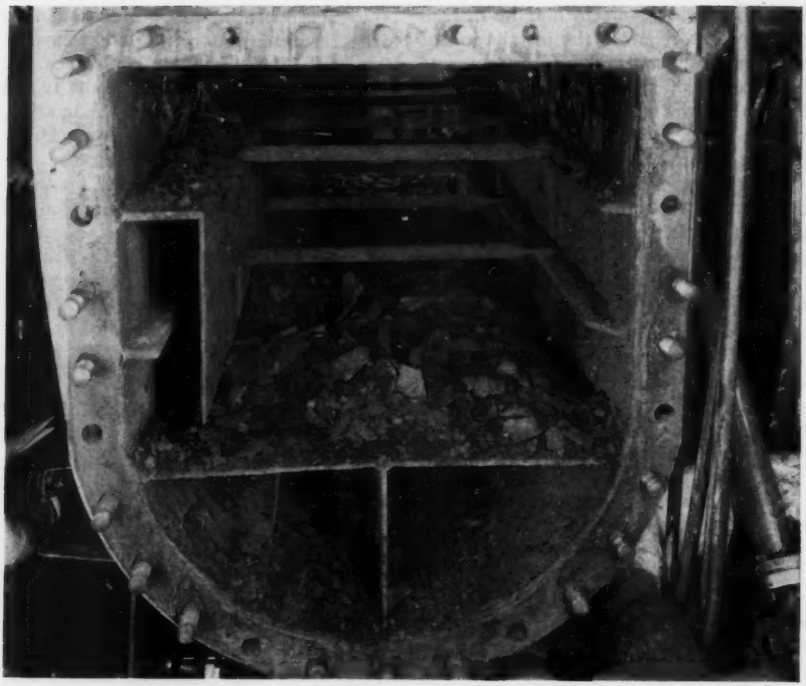


FIGURE 20.—Condition of Interior of Shell of Distilling Condenser—U.S.S. *Clark* (DD361).

condenser, inner feed heater and vapor feed heater were 20, 24, and 19, respectively, out of which 11.7 months constituted operation in Mediterranean waters.

*Evaporator Feed-Treatment System.*

During the summer of 1944, extensive tests to determine the value of pretreating evaporator feed with cornstarch and Navy boiler compound were conducted on distilling plants installed on board the destroyer tenders U.S.S. *Altair* (AD11) and U.S.S. *Hamul* (AD20). These tests were conducted during anchorage at Great Sound, Bermuda, under the direct supervision of Mr. Ray

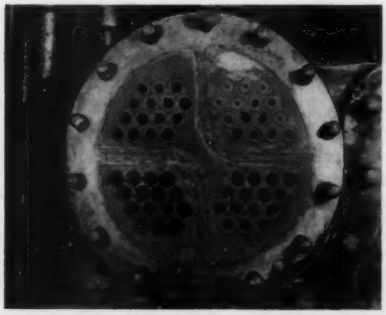


FIGURE 21.—Tube Fouling of Air-Ejector After Condenser — U.S.S. *Brooklyn* (CL40).

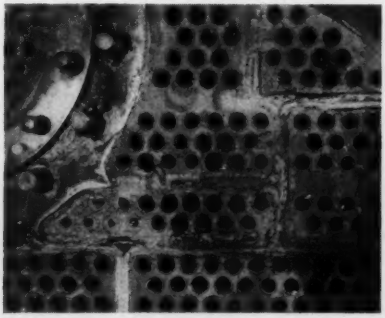


FIGURE 22.—Tube Fouling of Inner Feed Heater—U.S.S. *Brooklyn* (CL40)

FIGURE 23.—Tube Fouling of Vapor Feed Heater—U.S.S. *Brooklyn* (CL40).

L. Clapper, head of the Service Department of the Griscom-Russell Company. The sea water in Bermuda Harbor was found to be 1.25 thirty-seconds density and supersaturated with calcium carbonate. Before the tests were started, the distilling plants were cleaned thoroughly and placed in a satisfactory operating condition.

The results of these tests indicated that the capacity of the plant operating on straight sea-water feed was about equal to the plant operating with a pretreatment of 0.2 lb. cornstarch plus 0.2 lb. boiler compound per 1000 gal. up until 465,000 gal. of water were distilled, after which the capacity of the plant employing pretreatment of feed exceeded that of the plant operating on straight sea-water feed. However, the rate of heat transfer in gallons per hour divided by the temperature difference for both tests varied between 14.2 at the start and an average of 11.2 at the end of the production of 600,000 gal., equivalent to a 30-day period, thus resulting in a fairly flat rate of heat-transfer versus production characteristic.

Operation with untreated evaporator feed yielded a brittle, porous, and easily cracked scale; whereas, in the case of the plant operated with pretreated feed, there was no scale formation but instead an even coating of white powder was formed on the tube surface, principally on the first effect. The use of all boiler compound (0.4 lb. per 1000 gal.) resulted in the same conditions just outlined for the cornstarch-boiler compound pretreatment test. The use of all cornstarch (0.4 lb. per 1000 gal.) was not effective and normal capacity could not be maintained owing to rapid scale formation on all three tube nests. Also, the introduction of the feed-treatment compound through the tubes of the distilling condenser and vapor feed heaters was not effective and probably was responsible for the deposition of solids which clogged and heavily coated tube walls with paste material as well as hard scale. Figs. 24, 25, 26, and 27 illustrate the various types of scale and/or deposit accumulated during the aforementioned distilling-plant tests.

The Bureau of Ships, in the summer of 1944, authorized the installation of the cornstarch-boiler compound feed-treatment system for low-pressure distilling plants installed aboard all naval vessels in commission and under construction. Reports received from more than fifty vessels indicated the results were beyond all expectations. The mixture was injected into the bottom of the first-effect shell through a perforated pipe by means of "vacuum drag." The introduction of cornstarch-boiler compound not only resulted in the removal of existing scale but retarded scale formation during normal operation of the distilling plant. One destroyer reported that the distilling plant was in operation for nearly 18 months before the unit was shut down for cleaning. An auxiliary vessel reported that existing scale began to fall off after feed treatment was initiated and within a short period better than 80 per cent of the old scale had fallen off, and that the same quantity of water could be distilled in one half the time.

### *British Experience.*

Lieut. (E) A. L. Benke, RNR, in a paper before the Institute of Marine Engineers,[1] cited an interesting distilling-plant experience. The distilling plant consisted of two double-effect "soloshell" units installed aboard an American-built escort carrier, having a

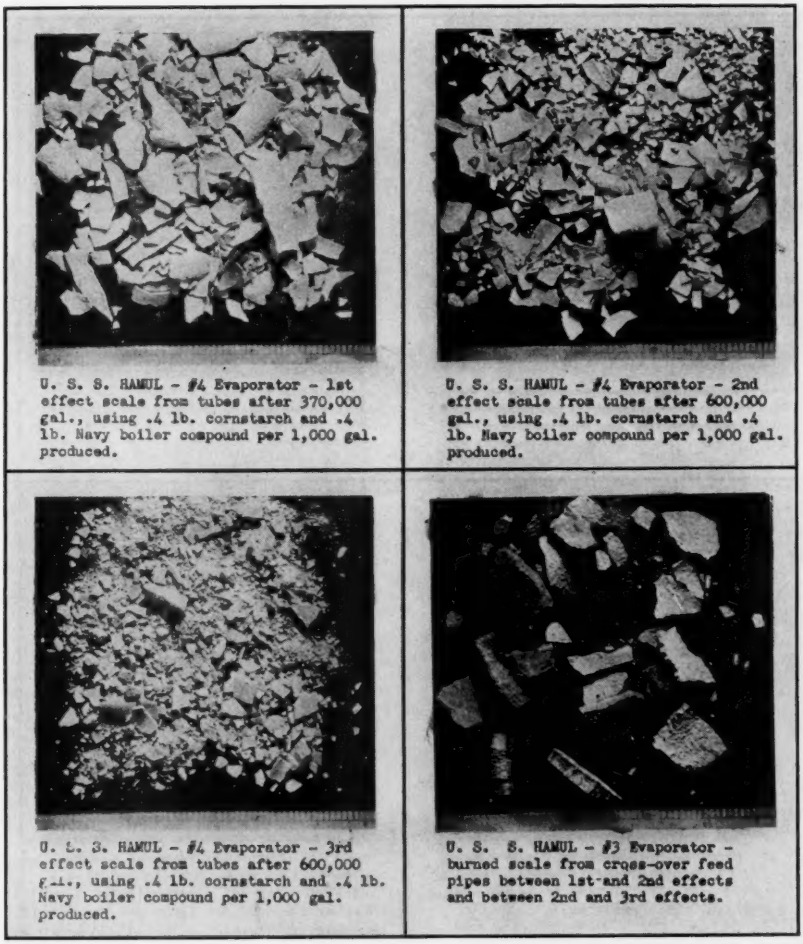
[1] "Starch Injection and Evaporator Output," by A. L. Benke, Proceedings of the Institute of Marine Engineering, vol. lvii, 1946, pp. 161-162.

U. S. S. ALTAIR - #1 Evaporator - 1st effect scale from tubes after 340,000 gal. No treatment used.
U. S. S. ALTAIR - #1 Evaporator - 1st eff. scale from tubes after 600,000 gal. No treatment used.
U. S. S. ALTAIR - #1 Evaporator - 2nd effect scale from tubes after 600,000 gal. No treatment used.
U. S. S. ALTAIR - #1 Evaporator 3rd effect scale from tubes after 600,000 gal. No treatment used.

FIGURE 24.—Samples of Scales Removed From No. 1 Evaporator—U.S.S. *Altair* (AD11). (Scale divisions at bottom of photograph = 0.1 in.).

normal designed output of 44 tons per day.

"In November, 1944, two Admiralty pattern starch-injection units were supplied and fitted, one to each evaporator. Some U. S. Navy boiler compound was added in solution with the starch, in the proportion of 4 lb. of starch and 1 lb. of compound to each evaporator daily. The result has been that since July, 1944, the tube nests have not required cleaning or scaling. Today there is the finest possible scale only on one effect coil. The ship left home waters on July 7, 1945, and has since operated in waters around the coasts of India, Ceylon, Malaya, Sumatra, and in the China Sea, in all of which waters, temperatures rarely fall below 82 deg. F., and seem to contain more than a normal quantity of scale-forming matter but due to the injection of the starch and boiler compound, none deposits on the tubes. As an indication of the efficiency of the plant operated in this way, a total of 7801 tons have been distilled during



U. S. S. HAMUL - #4 Evaporator - 1st effect scale from tubes after 370,000 gal., using .4 lb. cornstarch and .4 lb. Navy boiler compound per 1,000 gal. produced.

U. S. S. HAMUL - #4 Evaporator - 2nd effect scale from tubes after 600,000 gal., using .4 lb. cornstarch and .4 lb. Navy boiler compound per 1,000 gal. produced.

U. S. S. HAMUL - #4 Evaporator - 3rd effect scale from tubes after 600,000 gal., using .4 lb. cornstarch and .4 lb. Navy boiler compound per 1,000 gal. produced.

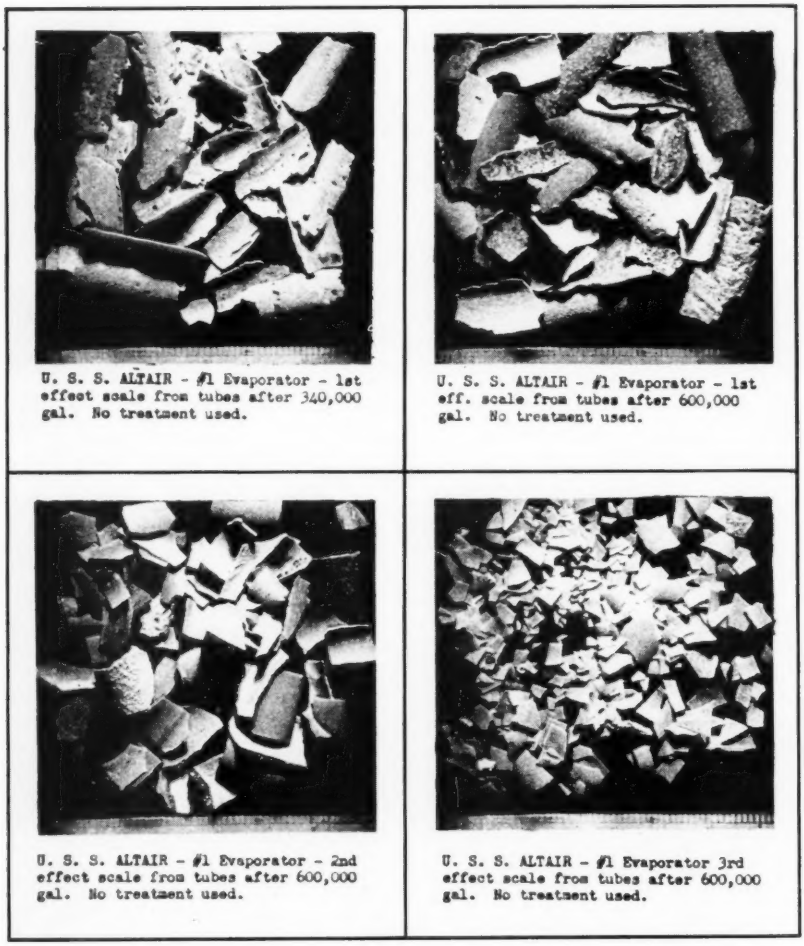
U. S. S. HAMUL - #3 Evaporator - burned scale from cross-over feed pipes between 1st and 2nd effects and between 2nd and 3rd effects.

FIGURE 25.—Samples of Scale Removed From No. 3 and No. 4 Evaporators—U.S.S. *Hamul* (AD20).

(Scale divisions at bottom of photograph = 0.1 in.).

an aggregate of 4878 working hours between July 1 and November 10, 1945, which gives an approximate daily average of 77 tons."

The British Admiralty has also experimented with the introduction of acid sodium sulphate and "Belloid T.D.," which is actually the sodium salt of a dinaphyl methane, disulphonic acid, and to date very satisfactory results have been obtained. Full-scale tests were made in standard evaporator equipment aboard the aircraft carrier H.M.S. *Implacable* over the past 2 years. Instead of preventing the deposition of scale, the aim is to control the physical form of the deposited matter so that hard and adherent scale is unlikely. The foregoing tests include comparisons between untreated sea water, sea water plus cornstarch-boiler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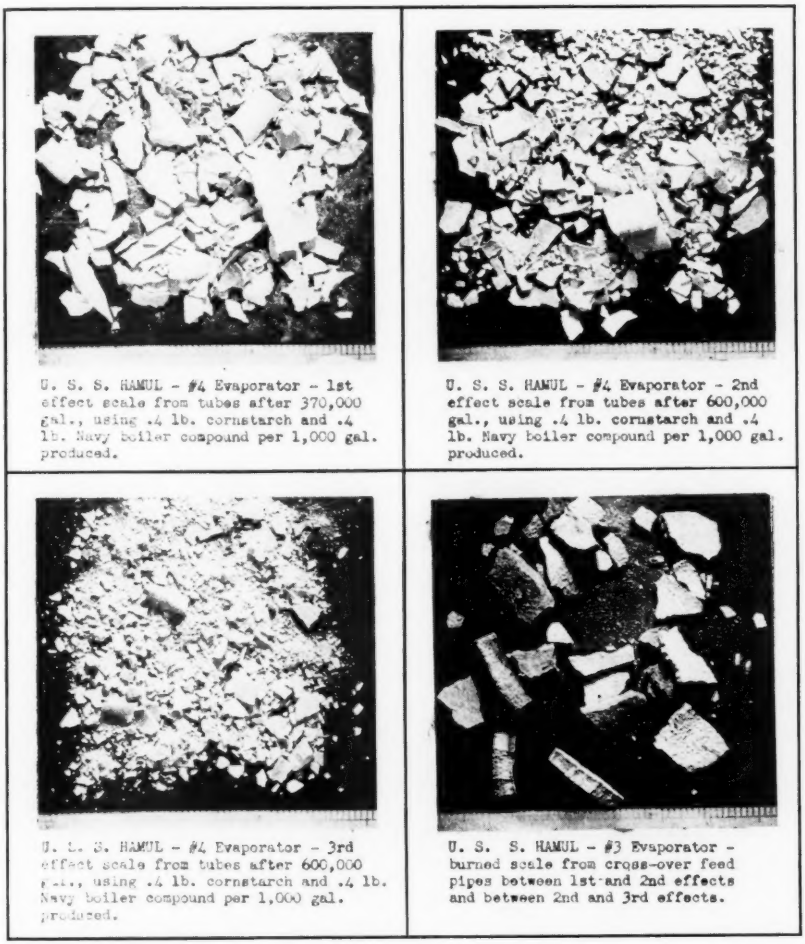

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U. S. S. HAMUL - #3 Evaporator - burned scale from cross-over feed pipes between 1st and 2nd effects and between 2nd and 3rd effects.

FIGURE 25.—Samples of Scale Removed From No. 3 and No. 4 Evaporators—U.S.S. *Hamul* (AD20).

(Scale divisions at bottom of photograph = 0.1 in.).

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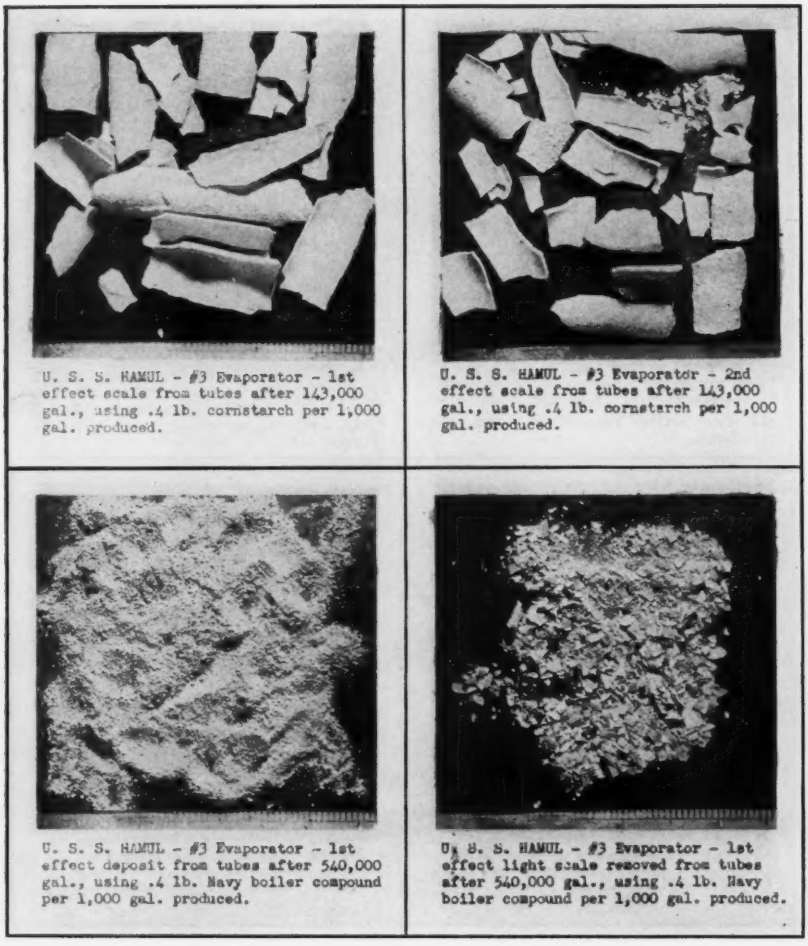


U. S. S. HAMUL - #3 Evaporator - 1st effect scale from tubes after 143,000 gal., using .4 lb. cornstarch per 1,000 gal. produced.

U. S. S. HAMUL - #3 Evaporator - 2nd effect scale from tubes after 143,000 gal., using .4 lb. cornstarch per 1,000 gal. produced.

U. S. S. HAMUL - #3 Evaporator - 1st effect deposit from tubes after 540,000 gal., using .4 lb. Navy boiler compound per 1,000 gal. produced.

U. S. S. HAMUL - #3 Evaporator - 1st effect light scale removed from tubes after 540,000 gal., using .4 lb. Navy boiler compound per 1,000 gal. produced.

FIGURE 26.—Samples of Scale Removed From No. 3 Evaporator—U.S.S. *Hamul* (AD20). (Scale divisions at bottom of photograph = 0.1 in.).

compound treatment and sea water plus various concentrations of "Belloid T.D." injections. The average feed treatment is 12 oz. for 10 tons of water distilled. The resultant scale is softer and less adherent, thus permitting it to be easily cracked from the tube surface under conditions of operation of the evaporator.

*Research Now in Progress.*

Considerable investigations into the causes and prevention of scale and corrosion in both low-pressure and vapor-compression types of distilling plants, employing sea water as the evaporator feed, are now in progress. The Navy, in collaboration with the Coast Guard, has placed a contract with Armour Research Foundation, whereas the Army has awarded a contract to the University of California for the purpose of trying to

Old scale removed from evaporator headers and from bottom of tubes - was formed when using cornstarch treatment.

Scale removed from inner surface of 1st effect vapor feed heater tubes - was formed during use of cornstarch treatment.

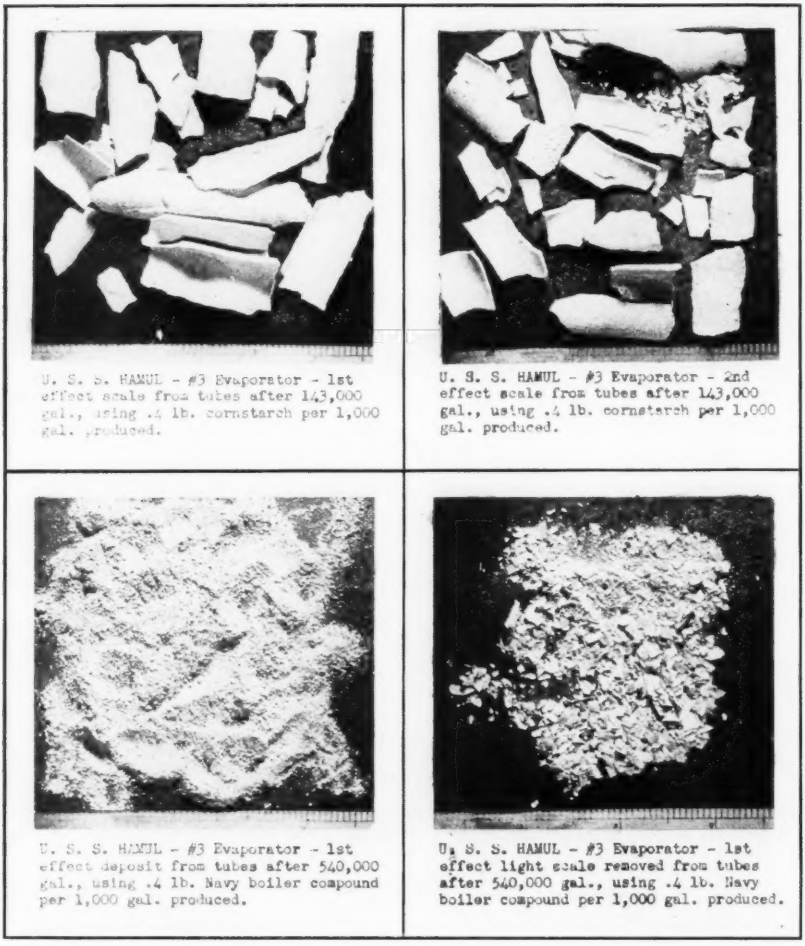
Scale deposit removed from feed heater tubes in paste form - accumulated during use of cornstarch treatment.

Vapor feed heater tubes solidly plugged with hard scale - formed during use of cornstarch treatment.

FIGURE 27.—Samples of Scale Removed From Auxiliary Units.
(Scale divisions at bottom of photograph = 0.1 in.).

find a satisfactory practical solution to this problem. The Bureau of Ships is also investigating other methods of scale prevention, such as a continuous ion-exchange system with a counterflow brine-regenerative arrangement, removal of carbon dioxide, and the installation of a specially designed electrolytic cell in the evaporator-feed circuit.

It is anticipated that good results will come out of the various research and development projects concerned with the prevention of evaporator scale, but, in the interim, the Bureau of Ships is requiring the design of vacuum-operated plants, both of the low-pressure and vapor-compression types, for new construction, in order to hold operating temperatures at a minimum.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U. S. S. HAMUL - #3 Evaporator - 1st effect deposit from tubes after 540,000 gal., using .4 lb. Navy boiler compound per 1,000 gal. produced.
U. S. S. HAMUL - #3 Evaporator - 1st effect light scale removed from tubes after 540,000 gal., using .4 lb. Navy boiler compound per 1,000 gal. produced.

FIGURE 26.—Samples of Scale Removed From No. 3 Evaporator—U.S.S. *Hamul* (AD20). (Scale divisions at bottom of photograph = 0.1 in.).

compound treatment and sea water plus various concentrations of "Belloid T.D." injections. The average feed treatment is 12 oz. for 10 tons of water distilled. The resultant scale is softer and less adherent, thus permitting it to be easily cracked from the tube surface under conditions of operation of the evaporator.

*Research Now in Progress.*

Considerable investigations into the causes and prevention of scale and corrosion in both low-pressure and vapor-compression types of distilling plants, employing sea water as the evaporator feed, are now in progress. The Navy, in collaboration with the Coast Guard, has placed a contract with Armour Research Foundation, whereas the Army has awarded a contract to the University of California for the purpose of trying to

Old scale removed from evaporator heaters and from bottom of tubes - was formed when using cornstarch treatment.
Scale removed from inner surface of 1st effect vapor feed heater tubes - was formed during use of cornstarch treatment.
Scale deposit removed from feed heater tubes in paste form - accumulated during use of cornstarch treatment.
Vapor feed heater tubes solidly plugged with hard scale - formed during use of cornstarch treatment.

FIGURE 27.—Samples of Scale Removed From Auxiliary Units.
(Scale divisions at bottom of photograph = 0.1 in.).

find a satisfactory practical solution to this problem. The Bureau of Ships is also investigating other methods of scale prevention, such as a continuous ion-exchange system with a counterflow brine-regenerative arrangement, removal of carbon dioxide, and the installation of a specially designed electrolytic cell in the evaporator-feed circuit.

It is anticipated that good results will come out of the various research and development projects concerned with the prevention of evaporator scale, but, in the interim, the Bureau of Ships is requiring the design of vacuum-operated plants, both of the low-pressure and vapor-compression types, for new construction, in order to hold operating temperatures at a minimum.

# NEW FACTORS TO BE CONSIDERED IN THE DESIGN AND WELDING OF SHIPS.

ACKNOWLEDGMENT.

This article, reprinted from the "Journal of the American Welding Society" for September 1948, is by MILTON FORMAN, of Battelle Memorial Institute. It was presented at the 29th annual meeting of The American Welding Society in Philadelphia, Pa. during the week of October 24, 1948.

The past fifteen years has been an unprecedented expansion in the application of welding. Not only has welding replaced riveting and casting in many fields, but a multitude of new products have been developed through the use of this highly versatile process.

The greatest progress has been achieved in the design and fabrication of small and medium sized structures. Here, welding has been tried and tested, and is accepted as the best and most economical process. Its application has especially been wide in the following industries: pressure vessels, machinery, sheet metal, ordnance, automotive, railroad, etc.

However, certain difficulties have been encountered in the design and construction of large all-welded structures such as bridges and ships. The highly publicized failures of T-2 Tankers and Liberty ships are still haunting the welding industry. The failures of these ships and some bridges acted as a catalyst for large scale research and scientific investigation into the characteristics of steel, the effect of welding and the standards of design for welding. These investigations have borne fruit and we are closer to a fuller understanding of the problem and its solution.

## *Riveted Ships Have Also Failed.*

It should be mentioned that structural failures were not limited to welded ships. E. M. MacCutcheon has reported that the *Oakley L. Alexander,* the *Oklahoma* and the *Mielero,* all riveted ships, broke in two in a manner practically identical to the failures of the welded ships. In addition, it is known that the *Leviathan* and the *Majestic* cracked their main decks. There is no doubt that there were many less spectacular fractures on riveted ships which have never been recorded. It has only been since the failure of welded ships built in World War II that complete records have been kept.

### *Approach of Designer.*

In the past, a naval architect, in designing a ship, was limited by a rather static point of view. He estimated the load and computed the stress assuming that each component part would carry its proportion of the load. This was assumed regardless of the notch or discontinuity inherent in the design. That is why hatch corners were designed square, and felt to be safe if the scantlings were sufficient. It has long been known, and sad experience has again taught us that sharp discontinuities concentrate stress in a relatively small area, and that this concentration of stress can exceed the fracture strength of the material in way of the notch. Thus, the structure can fail at a load far below its design strength. This is especially true if local elongation does not occur near the notch and thereby prevent a redistribution of stresses to other locations away from the root of the notch. This basic fact holds true whether the structure is riveted or welded. The reason why riveted structures do not seem to be as susceptible to notch effects is that a certain amount of elongation and slippage takes place at the riveted joints and the stress concentration is reduced at the notch itself. The general behavior is the same, the degree of stress concentration and the elongation of the specimen as a whole is different.

### *Difference Between Riveted and Welded Structures.*

As we all know, the very act of welding develops a continuity which had not previously existed in riveting. This continuity results in a different set of conditions for the transmission of stresses when a load is applied to the structure. Fig. 1 portrays a simplified version of what happens. When the unnotched riveted specimen No. 1 is pulled in tension, there will be a concentration of stress at the many rivets, and as the load is increased, slippage will occur at the rivets. If the rivet size and spacing are correct, the specimen will fail only after its full design strength has been utilized. The specimen will behave in a ductile manner and the fracture will be shear in character.

In the case of unnotched welded specimen No. 2, the full design strength will also be achieved. However, there will be nothing to impede the uniform distribution of stress throughout the entire section. The specimen will also act in a ductile manner and a similar shear fracture will result. It is therefore evident that unnotched specimens, both welded and riveted, will develop their full design strength. Of course, the riveted specimen would either have to be wider or thicker in order to sustain a load equal to that of the welded one.

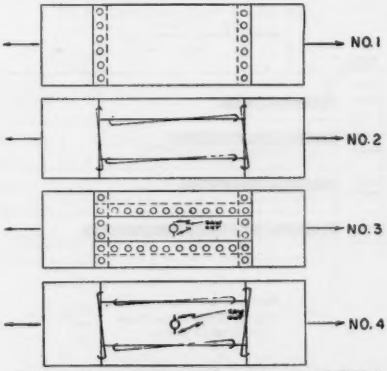



Fig. 1—Specimens Showing Behavior of Riveted and Welded Structures When Loaded

However, when the notched specimens Nos. 3 and 4 are tested, it will almost be impossible to predict their point of failure by calculating the effective cross-sectional area. The riveted specimen No. 3 will undoubtedly absorb more energy before failure. For, when the

load is applied, a redistribution of stress will take place at the rivets, and therefore the full concentration of stress will not immediately be developed at the notch located in the center of the plate. In specimen No. 4, the stress will be uniformly distributed except in the area of the notch. There will be a greater concentration of stress at the notch with no dissipation at the rivet holes. In addition, *the notch in the welded specimen may be more effective because of greater constraint or "size" effect.* The mass of restraining material in the area of the notch is greater in the continuous, welded specimen than in the discontinuous riveted one. On the whole, the riveted specimen will act in a more ductile manner.

## *Importance of Design.*

The research of the past few years has tended to concentrate attention on notch sensitivity as the basic cause of ship fractures. While it is true that the low notch toughness of ship steel is a precondition for premature failure, I believe more emphasis can be placed on the design aspects of the problem. Recent work done at the University of California and statistical records of the U. S. Coast Guard seems to bear this out.

## *Effect of Design on Hatch Corners.*

Fig. 2 charts some of the results of hatch corner tests at the University of California. It is to be noted that in each case the same notch sensitive ship steel was used, and the welding electrode and

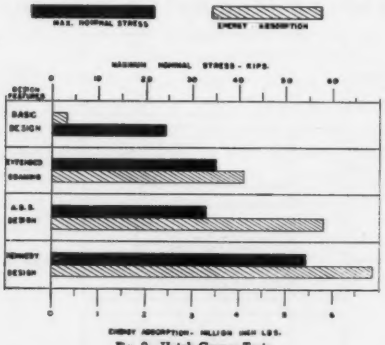



Fig. 2—Hatch Corner Tests
From De Garmo

welding procedure was standard. *A careful analysis of the results shows that any design change which tended to alter the flow of stress from the square hatch corner itself brought greatly improved results.* The basic square hatch corner is shown in Fig. 3. The only improvement in Fig. 4 is that the longitudinal coaming was extended for one frame space to

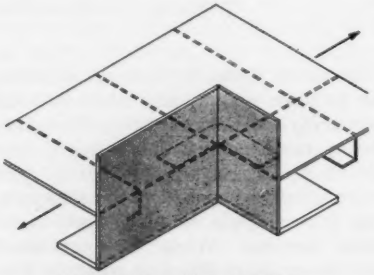


Fig. 3—Hatch Corner Tests. Basic Design
From De Garmo

attach to the deck. This allowed a fair proportion of the load to be carried to the deck itself instead of concentrating it at the corner.

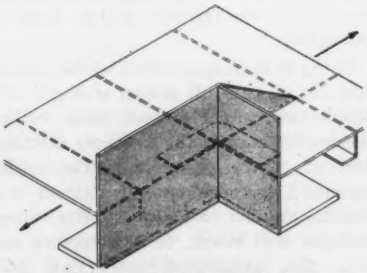


Fig. 4—Hatch Corner Tests. Extended Coaming Design
From De Garmo

The A.S.B. design in Fig. 5 is a combined rounded and square design. It is to be seen that it also shows decided improvement over the basic design. However, the maximum nominal stress as shown in Fig. 2 is slightly below the extended coaming design, while the energy absorption is considerably improved. In essence, both Figs. 4 and 5 remain square hatch corners with alterations to avoid the full concentration of stress at the square discontinuity The extended coaming seems most effective in increasing the breaking stress while the rounded off deck and girder flange of Fig. 5 seems to contribute most to increasing ductility and energy absorption.

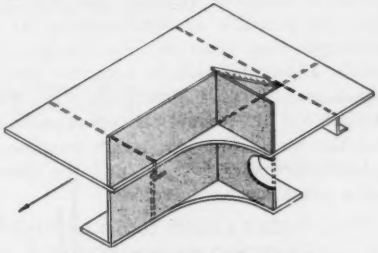


Fig. 5—Hatch Corner Tests. A.B.S. Design From De Garmo

*Kennedy Design Is Best.*

Fig. 6 shows the ideal Kennedy design. Here, for the first time, all aspects of a square hatch corner have been eliminated. A smooth transition has been achieved from every point of view. The test results in Fig. 2 show that the maximum nominal stress and the energy absorption is now within close range of the standard 0.505-in. tensile specimen. Therefore, despite the inherent design discontinuity, almost ideal conditions have been achieved for a smooth transmission of stresses.

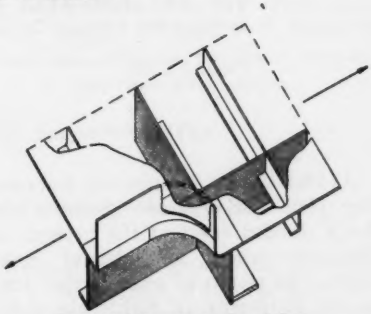


Fig. 6—Hatch Corner Tests. Kennedy Design From De Garmo

*Test Correlation with Actual Service.*

E. M. MacCutcheon has reported that the University of California test results conform well with the actual service of these hatch corners on ships. (It is to be noted that the Kennedy design has not as yet been used in ship construction.) Mr. MacCutcheon's discussion also points out that the hatch corner shown in Fig. 7 has had 7722 ship months' service time and no fractures have ever been reported. The test results at the University of California of this hatch corner showed that the maximum nominal stress and energy absorption was about equal to the results shown in Fig. 2 for the extended coaming design. Since the correlation with actual service is favorable, it should be possible, within limits, to predict the ship service of various types of hatch corners on the basis of the full-scale tests. As such, there should therefore be good reason to accept the Kennedy design as ideal for ship design.

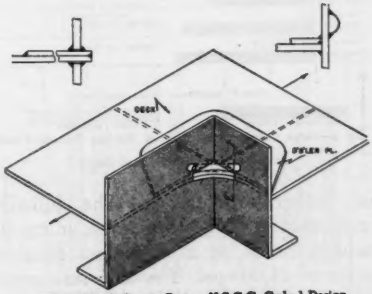


Fig. 7—Hatch Corner Tests. N.S.C.G. Code 1 Design From De Garmo

*General Comments.*

It is to be noted that contrary to the other designs, the Kennedy type does not use the principle of reinforcement as a strengthening measure. The design, as such, allows for ductile behavior under load and permits practically the full utilization of the physical test properties of the steel. The low notch toughness plays practically no role since the effect of a notch has been avoided.

Fig. 8 shows a hatch corner, developed at Ingalls during the war, and used on C-3 Hulls. Although I do not have complete service time records, it was used on many ships which have been operating for years. To my knowledge, no fractures have ever been developed at it. Although it is superior to most other designs, it is my opinion that it is still inferior to the ideal Kennedy design.

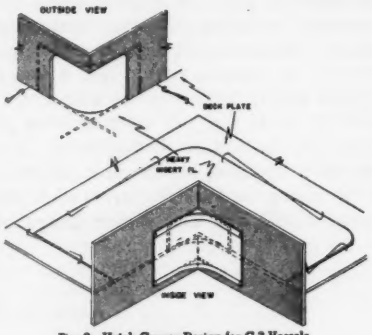



Fig. 8—Hatch Corner Design for C-3 Vessels Developed at Ingalls Ship Corp.

*Effect of Size and Thickness on Design Strength.*

Available test information indicates that plate thickness and specimen size has a decided effect of the strength of structures where an inherent notch exists. The results of some further tests at the University of California makes available interesting information on this subject. Part of the test results of Fig. 9

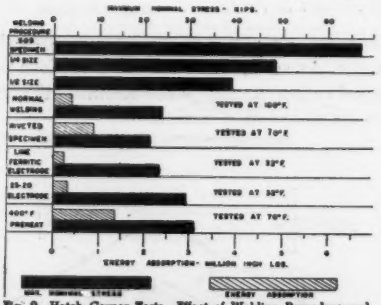



Fig. 9—Hatch Corner Tests. Effect of Welding Procedure and Size
From De Garmo, Merrian and Grassi

are rather illuminating. The full-size basic square design specimen, normally welded, broke at a maximum nominal stress of 24,000 psi. The half size specimen reached a maximum stress of 39,000 psi., while the quarter size specimen reached as high as 48,000 psi. The effect of specimen size, therefore, had a major effect, since this was the only variable in these tests. Some other interesting test results are shown in Fig. 10 on notched tensile specimens of various diameters. It is to be noted that both the notch strengh and notch ductility are greatly reduced as the specimen diameters are increased from ¼ to 1½ in.

The thickness of the plate seems to have a distinct relationship to the notch toughness of the resultant structure.

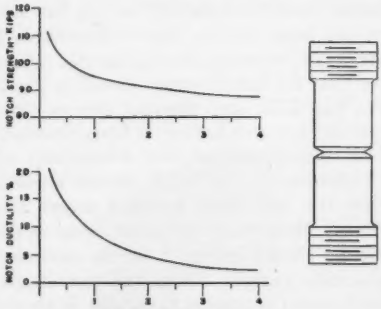



Fig. 10—Effect of Size on Notched Tensile Specimen
From Brown, Lubahn and Ebert

Fig. 11 supplies additional evidence in support of this assertion. The same steel was tested under the same conditions except that some of the specimens were ½ in. in thickness and the others were 1 in. The specimens were tested by the slow-bend notched-bead test. It is to be noticed that the bend angle is approximately ⅛ in the case of the thicker 1-in. plate. One could conclude from this that the effectiveness of the notch is dependent on the mass of restraining material in the immediate area.

The above results may indicate a new approach to designing in way of discontinuities. *It may be advisable to make every effort to reduce the thickness of plating in way of discontinuities, instead of increasing the same through the use of doubler plates.* The above may possibly be achieved, either through the use of higher strength steels in way of discontinuities, or by designing the structure so that the heavier members are located in those portions which contain no change in continuity.

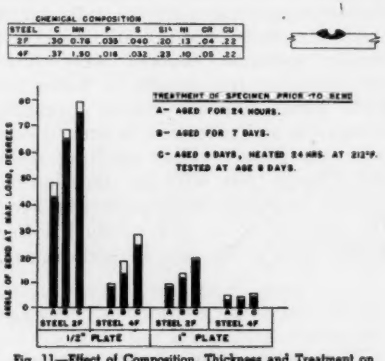

CHEMICAL COMPOSITION
STEEL C MN P S SI% NI CR CU
2F .30 0.78 .035 .040 .20 .13 .04 .22
4F .37 1.50 .016 .032 .23 .10 .05 .22
TREATMENT OF SPECIMEN PRIOR TO BEND
A- AGED FOR 24 HOURS.
B- AGED FOR 7 DAYS.
C- AGED 6 DAYS, HEATED 24 HRS. AT 212°F.
TESTED AT AGE 8 DAYS.
ANGLE OF BEND AT MAX. LOAD, DEGREES
A B C
STEEL 2F
STEEL 4F
1/2" PLATE
1" PLATE

Fig. 11—Effect of Composition, Thickness and Treatment on Welded Specimen
From Voldrich, Bennett and Martin

*Box Girder Tests at National Bureau of Standards.*

Five large 9 ton girders were built for testing at the National Bureau of Standards. Each was 23 ft. 6 in. long and constructed from 1½- and 2½-in. plate as shown in Fig. 12. The girders were designed and welded in order to develop the most severe restraint through an abusive welding sequence and excessive welding. The aims of the tests were to determine the effect of testing temperature and severe geometrical constraint on a large rigid structure. Care was taken that the design and workmanship did not introduce a deliberate notch effect. The welding sequence was executed to develop as high reaction and residual stresses as possible.

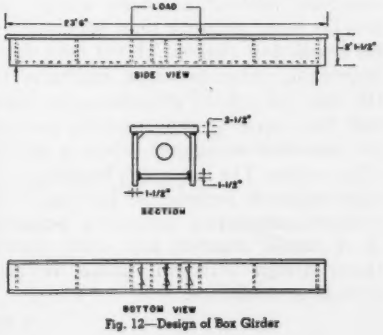



Fig. 12—Design of Box Girder

An examination of the physical properties as measured by tensile specimens are recorded in Fig. 13. A glance will

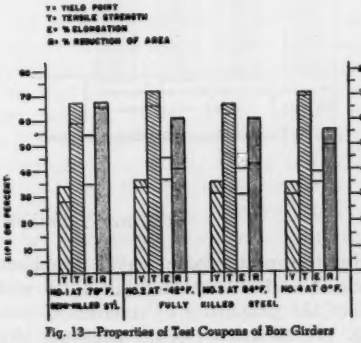



Fig. 13—Properties of Test Coupons of Box Girders

show that the semikilled, ship steel of Box Girder No. 1 did not vary much from the fully killed steel of the other girders. In fact, the semikilled steel had better elongation and reduction in area than the fully killed steel. Also, various testing temperatures of the fully killed steel, as measured by an unnotched tensile specimen, showed very little difference in test results in terms of yield point, ultimate tensile strength, elongation and reduction in area. However, the impact values resulting from the Charpy test with an Izod notch, showed very great discrepancies between the two steels, as well as much lower values with a decrease in temperature. The semikilled steel had much lower impact values at room temperature (Fig. 14).

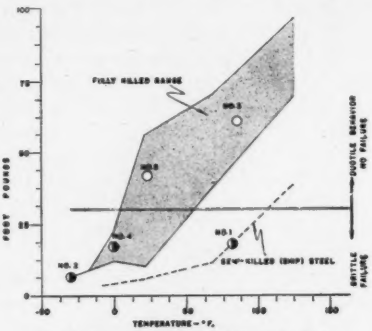



Fig. 14—Charpy Impact Values of Box Girder Steel From Boyd

*Test Results.*

There seems to be a rough correlation between the impact values and the test results. When the impact values were below 25 ft.-lb. at a specific testing temperature, the girders broke in a brittle manner. When the impact values were above 25 ft.-lb., the girders behaved in a completely ductile manner and did not break at all. As shown in Fig. 15, fully killed Box Girder No. 4 tested at 0° F. resisted a greater load than semikilled Girder No. 1 at 78° F.

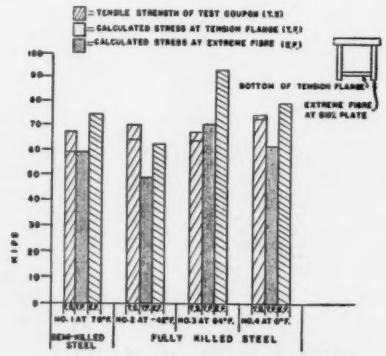

= TENSILE STRENGTH OF TEST COUPON (T.S)
= CALCULATED STRESS AT TENSION FLANGE (T.F)
= CALCULATED STRESS AT EXTREME FIBRE (E.F.)
BOTTOM OF TENSION FLANGE
EXTREME FIBRE AT BOG PLATE
KI/SQ
NO. 1 AT 78°F.
NO. 2 AT -42°F.
NO. 3 AT 64°F.
NO. 4 AT 0°F.
SEMI-KILLED STEEL
FULLY KILLED STEEL

Fig. 15—Calculated Stress in Box Girders at Failure

It is important to recognize that despite the low impact values of the steel itself, 4 of the 5 girders, developed more than their calculated design strength at the extreme fibers of the side plate. Even in the case of Box Girder No. 2, tested at —42° F., the calculated stress at the extreme fiber was 60,000 psi. as compared to the ultimate tensile strength of the test coupon of 70,000 psi., at the same temperature.

It would therefore seem that a *welded structure without a design notch can develop more strength than indicated by the tensile test coupon.* This is even more important, since previous emphasis in the case of welded structures has been that they never seem to perform as well as unwelded structures when a notch effect exists. The low notch toughness of steel tended to becloud the fact that well designed unnotched structures perform in a ductile manner and easily fulfill their design strength under normal operating conditions.

*Greater Familiarity with Steel Needed.*

In the past, some naval architects seemed to be concerned but little with all of the properties of steel or its testing. We all know hat the accepted physical tests of steel to determine whether it meets specifications only use ideal, unnotched conditions. Since we were not too well aware of the importance of

notch sensitivity, little effort was put into eliminating those design features which produced a notch effect. The research and thinkng of the past few years has placed great emphasis on this very important characteristic of steel. Notch toughness is not at all a static property of steel. Notch toughness it not at all a static property of steel. It varies much more than any of the other properties of steel when influenced by any single one of the following factors; sharpness of the notch, rate of loading, temperature, deoxidation practice in the making of the steel, metallurgical structure.

### *Sharpness of Notch.*

It is rather obvious that the sharper the discontinuity or notch effect, the greater will be the concentration of stress at the root of the notch. In addition, the sharper the notch, the more it will tend to restrict plastic, ductile behavior and cause the structure to fail at a lower load and in a brittle manner. Laboratory evidence substantiating the above has been presented by Zeno and Low. Their tests show that the sharper the notch, the higher is the transition temperature to brittle fracture. Also, the deeper the notch for a given specimen thickness, the lower is the transition temperature. The latter again emphasizes the "size" effect, since the deeper the notch, the less is the mass of restraining material and the less is the constraint.

### *Rate of Loading.*

It has been fairly well established that the more rapid the loading rate on a notched specimen, the lower will be its resistance to fracture. The rate of loading, however, becomes a factor when its value is well above those produced by static loading. In the case of ships, the loading rates are rather low and fall within the general range of static loading. From all information available to this author, in no case has it been established that the loading rates encountered in ship operation reach a point where they would be considered a significant factor.

### *Temperature.*

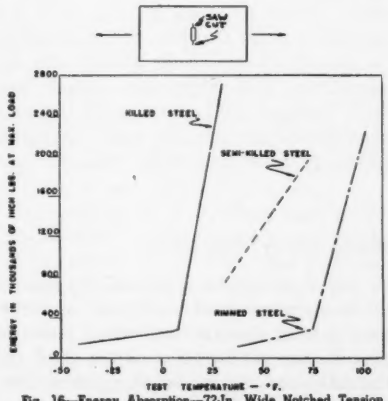



Fig. 16—Energy Absorption—72-In. Wide Notched Tension Specimen
From H. Hiemke

The lower the temperature, the less is steel able to absorb energy, and the lower is its notch toughness. This has been rather well established by such studies as shown in Fig. 16. It is readily evident from this that slight changes in temperature, such as 10 or 15° F., reduce the ability of steel to absorb energy to an alarmingly low point. When tested by notched tensile specimens, it is noted that rimmed steel is extremely low in energy absorption at room temperature, while killed steel reaches its low values at + 10° F. The above results are confirmed by many tests including the standard impact and slow bend tests.

### *Deoxidation Practice.*

Deoxidation practice in the steel mill has long been known to have a decisive effect on the impact values and notch toughness of steel. Another glance at Fig. 16 confirms this when the semikilled and fully killed steels are compared. At 25° F., killed steel absorbed almost 2,800,000 in.-lb., while the semikilled steel absorbed less than 800,000 in.-lb.

### *Heat Treatment and Metallurgical Structure.*

The heat treatment and metallurgical structure of steel has a decided influence on the notch toughness of tested specimens. Fig. 17 depicts one such comparison between a single type of steel in the as-rolled condition as compared with the spheroidized heat-treated condition. It is to be seen that the bend angle of a slow-bend notched specimen shows much better bend angles for the spheroidized specimen than for the as-rolled one.

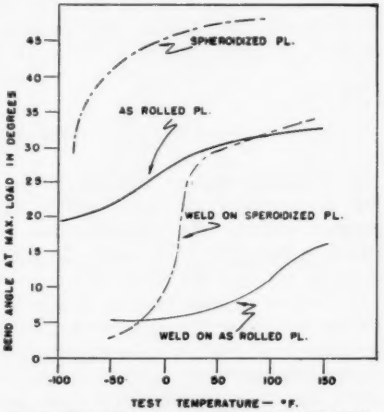



Fig. 17—Effect of Pretreatment on Welded Silicon Killed-0.25 Carbon Plate
From Stout and McGeady

### *Effect of Welding.*

A great deal of evidence is available showing that the welding process has little if any effect on an unnotched tensile specimen in terms of yield point, ultimate strength, reduction in area and percent elongation. This fact previously lulled us into a feeling of security. However a preponderant amount of evidence now points to the conclusion that welding has a negative effect on the notch toughness of steel.

Another glance at Fig. 17 shows the effect of welding on both the as-rolled and the spheroidized plates. It will be noted that, in these notched, slow-bend specimens, the bend angle in the as-rolled plate at 0° F. was reduced from 27 to 5½°. The spheroidized plate suffered similar results with the bend angle being reduced from 45 to 10° at the same temperature.

This detrimental effect of welding on notched specimens has also been confirmed by all other tests which attempt to evaluate notch toughness and notch ductility. It must be emphasized that a sharp notch or discontinuity must be present in order for this effect of welding to become operative. It should also be considered that structures, which are well designed and constructed through the elimination of any type of notch effect, perform extremely well.

### *Welding Procedures Can Improve Performance.*

The performance and strength of a welded structure can be improved by the use of different types of electrodes, preheating, postheating and special welding procedures.

*Welding Electrodes*: Referring back to Fig. 9, it will be seen that the use of 25-20 stainless steel electrodes and the lime coated ferritic electrodes tend to give better performance when used on the full-sized, square hatch corners. The two hatch corners, welded with the above

two electrodes, were tested at a temperature of 68° F. below the normally welded one. Yet, the lime-coated, ferritic-welded one shows about the same strength, and the 25-20 stainless steel welded one shows an increase of approximately 5000 psi.

The above is confirmed by an analysis of the transition temperatures to brittle fracture of various types of electrodes and welding methods portrayed in Fig. 18. It is immediately evident that the lime-coated ferritic electrode and the process of automatic welding gave very low transition temperatures in the weld metal as compared to the unaffected base plate. The highest transition temperature is encountered about 1 in. from the center of the weld in the heat-affected zone. At that, the highest transition temperature at this point is never more than 13° F. above that of the unaffected plate.

The use of E-6020 electrodes shows better results than E-6010. The E-6010 electrode shows its worst effect at the fusion zone, while the E-6020 reaches the highest transition temperature well in the heat-affected zone.

It is rather interesting to note that the oxyacetylene welding process offers the poorest results, and the zone most affected is the center of the weld itself.

It is to be remembered that, in the testing of notched specimens, the lower the transition temperature to brittle fracture, the better is the notch toughness of the steel.

Many reasons have been advanced for the diverse effect of different types of electrodes and processes on the notch toughness of steel. Some of these reasons include: heat input and its effect on the metallurgical structure due to a change in the cooling rate, chemical composition and hydrogen content of the weld metal and, to a much smaller extent, stresses due to the weldng process itself.

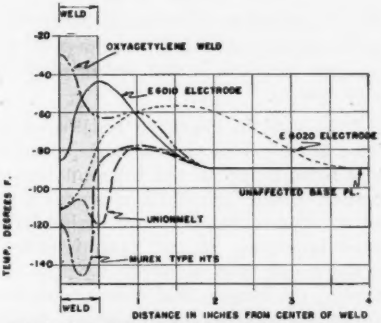



Fig. 18—Effect of Welding Technique on Transition Temperature
From Grossman and Shepler

*Heat Input Affects Notch Toughness.*

Heat input, as measured by welding speed where other factors are constant, seems to have a direct effect on notch toughness. Fig. 19 amply shows that a welding speed of 6 in. per minute results in higher bend angles for notched, slow-bend tests than a welding speed of 10 in. per minute. This holds true both for semikilled and fully killed steels.

In addition, Fig. 9 shows that a preheat of 400° F., gave much better results in the hatch corner tests. Not only was the ultimate unit strength increased by 7000 psi., but the energy absorption increased from 400,000 in.-lb. to 1,350,000 in.-lb. It seems here that the arresting of the cooling rate and the improved resultant metallurgical structure contributed to the better results.

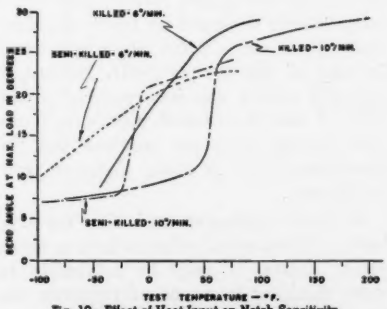



Fig. 19—Effect of Heat Input on Notch Sensitivity
From Stout, McGeady, Sun, Libsch and Doan

### *Aging Improves the Notch Toughness of Welded Specimens.*

Some of the work at Battelle offers an explanation for a phenomenon noticed in weld testing. We are all familiar with the fact that the incidence of failure of weld bend specimens seems to be higher when the specimen is bent immediately after welding. Fig. 11 depicts this rather well. It can be seen that the specimens which were aged 7 days gave better bend angles in the notched, slow-bend test. This might well be remembered when welder qualification tests are conducted.

### *Practical Conclusions Relating to Ship Construction.*

1. It has been rather well established that square hatch corners can fail at relatively low stresses in the neighborhood of 24,000 psi. Therefore this type of discontinuity should be avoided at all possible locations in the development of ship design.

2. Most of the improved designs, which have already been applied to ships, maintain some aspect of the inherent square corner. Some individual characteristics of the square hatch corner design have been improved. In one case, improved performance was achieved by only extending the longitudinal coaming for one frame space, retaining the square hatch corner as such. In all of the altered designs, improvement is achieved because more favorable conditions were developed for the transmission of stresses by the avoidance of the full concentration of stresses at the square corner itself. In all cases, except the Kennedy design, a reinforcing doubler plate is proposed.

3. The most effective new design presented completely eliminates all characteristics of the square corner. This is the hatch corner designed by Harry E. Kennedy. Here, no doubler plate was fitted in way of the corner itself. Instead, a rounded corner was incorporated in the vertical and horizontal members. Complete ductile behavior resulted and the specimen failed at some distance from the corner.

4. Since thickness and size seem to have a detrimental effect where a notch effect exists, it may be advisable to start thinking in terms of reducing the thickness in way of discontinuities and compensating for it, by either the use of higher strength steels or by using increased sections in way of neighboring locations which do not possess discontinuities.

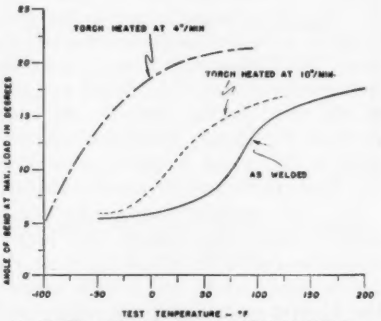


Fig. 20—Effect of Torch Postheating on Welded 0.25 Carbon-Silicon Killed Steel
From Stout and Mc Geady

5. Riveted structures with discontinuities do not show greater ultimate unit strength than welded ones. But they do tend to absorb more energy. There is no basic difference between the behavior of riveted and welded structures under conditions of a notch effect.

6. Any of the above-mentioned changes in hatch corner design had a greater effect on improving their performance than any change in welding electrode, preheat, postheat and welding procedure. Therefore, regardless of the improvement of the notch toughness of steel, the effect of the notch must be eliminated from design and workmanship.

7. Semikilled ship steel has rather low notch toughness and ductility at room temperature. Efforts are being made to

change the qualities of this steel in this respect. In addition, specifications, at present, do not call for a test to evaluate the notch toughness of steel. Many small tests are being considered. The general trend is in the direction of a slow-bend ductility test along the lines of the one proposed by Dr. A. B. Kinzel in Fig. 21. Evidence is available which indicates that this type of test would be able to fairly closely evaluate the ductility and resultant behavior of the steel structure under service conditions.

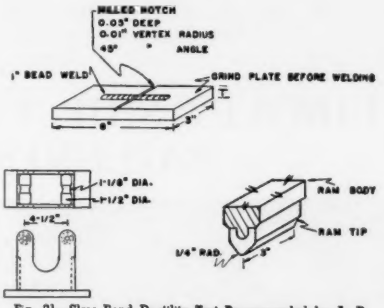



Fig. 21—Slow Bend Ductility Test Recommended by A. B. Kinzel

8. The act of welding to a certain extent impairs the notch toughness of steel. Methods are available which can restore this property to the steel. The most effective methods are preheating and postheating. Different types of electrodes also have some little effect on improving notch ductility.

# LIBERTY SHIP TAILSHAFT FAILURES.

ACKNOWLEDGMENT.

The following article from "The Marine Engineer" for September 1948 describes the modification of a Liberty Ship tailshaft to eliminate the cause of failures which have been experienced in this design as a result of torsional vibration.

Frequent machinery failures in Liberty-type vessels have necessitated intensive study of their design and practical operation to discover the causes of such trouble and to determine and apply the necessary corrective measures. Reasons for these failures, as proved by actual physical condition and theory, centre around the vibratory torsional stresses set up in the shafting system at critical revolutions, especially if the main engine is operated at or near its maximum designed speed for a sufficient period of time. The continuation of shafting troubles in these wartime ships has risen to such proportions that the American Bureau of Shipping have instructed their surveyors to reduce their tailshaft-drawing dates from every three years to a maximum of two years for Liberty-type vessels. The most prevalent appearance of this failure is a fracture in the large end of the taper of the tailshaft, in way of the keyway. To discover this fracture, the tailshaft must be drawn and submitted to a magnetic particle (Magnaflux) examination, as normal inspection methods are not sufficiently positive. The Todd Shipyards Corporation advise us that recent tailshaft examination of the Liberty ships *Samuel Colt, John Goode* and *Despina,* in their yard, disclosed this particular form of failure, new tailshafts being installed in each case.

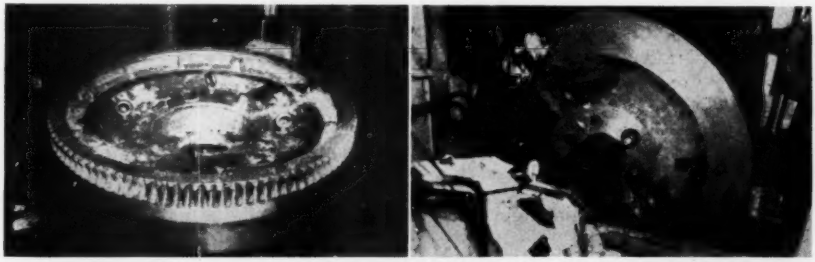
The American Bureau of Shipping have recommended to the owners of ships of this type to limit their propeller shaft to 66 r.p.m. for safe operating conditions, although the designed speed is 76 r.p.m. Records of actual tailshaft failures gathered by various owners and interested surveyors and inspectors have been available for study. Various tests have been conducted and four modifications to overcome such failures have been recommended by the American Bureau of Shipping, one of which is to install a flywheel aft of the main engine. This lowers the natural vibratory frequency of the system sufficiently to permit the operation of the engine at the normal maximum designed speed of 76 r.p.m.

### *Flywheel installation proposed.*

In anticipation of the action taken by the classification society in order to maintain the designed speed of the Liberty ships, Todd Shipyards Corporation authorized Gibbs & Cox, the well-known New York naval architects, to make a complete study and prepare plans for the approval of the American Bureau of Shipping for these various recommendations. The first recommendation

to be used was the installation of the flywheel installed on the main engine line shaft at the coupling between No. 6 main bearing and thrust bearing. The size and weight, determined by calculations from torsiograph curves, was subsequently approved by the American Bureau of Shipping and the U. S. Coast Guards.

*The John Goode's installation.*



The completed new flywheel is shown in the Todd shops on the left and in the other view fitted in place in the Liberty-type tanker *John Goode.* Note the new position of the steam-driven turning gear.

Todd's Los Angles yard recently fitted such a flywheel to the *John Goode,* a Liberty tanker purchased a short time ago by World Wide Tankers, Inc., of Los Angeles and Seattle. The flywheel was duly installed using drawings furnished by Gibbs & Cox, the modification complying with the regulations and passing the inspection of the American Bureau of Shipping and the U. S. Coast Guard. As there are so many Liberty ships afloat some notes on the work will be of wide interest to readers all over the world.

*Castings used.*

The important parts being castings, patterns were made for the following: flywheel, worm wheel gear, and turning gear pedestal. The rough weight of these castings aggregated 19,930 lb., including $\frac{3}{8}$ in. material allowance for finish machining. The flywheel, having been cast in halves, required facing and bolting together prior to being rough- and finished-machined in a Cincinnati vertical boring mill.

The new worm wheel for the modified position of the turning gear was cast in four 90 deg. sections. This necessitated facing each section prior to assembly and bolting. The assembled gear was then laid out and formed in order to machine to diametrical pitch. After being bored to fit the flywheel, the gear was fastened to it with bolts. The finished weight of the complete flywheel (wheel and gear) was 15,005 lb. The only requirement for special tools was the wrench for fastening the flywheel halves together at the nut core near the periphery of the wheel. No special jigs or fixtures were required for the machinery operations.

While the flywheel and associated parts were being machined in the shop, the existing turning gear, shaft coupling bolts, No. 6 bearing cap piping and gratings were removed. The completed flywheel was disassembled and lowered into place on the ship in halves. After rebolting and aligning with the coupling

bolt holes, the wheel was bored for coupling bolts.

The No. 6 bearing cap was refitted and the turning engine was installed in its new position. After aligning the worm gear of the turning engine with the worm wheel, the pedestal foundation was drilled for fitted bolts. With the foundation secured, all alignments were checked by rotating the flywheel. Associated equipment such as piping, gratings and guards were then reinstalled.

*Torsiograph checks.*

Instruments to record on paper the torsional vibrations of a rotating shaft (Geiger & Cox torsiograph instruments) were installed and checked during a brief dock trial held on July 22 last. These machines were placed in three separate locations, namely:

1. Forward of the main engine on the H.P. crankshaft.
2. Aft of the thrust bearing, and
3. Forward of the stern tube but aft of the spring bearing.

These separate locations permitted (1) comparison of the recordings, (2) accurate analysis of vibration, and (3) calculations for stress curves. Three records were imposed upon a tape simultaneously during the operation of these machines, namely: Actual vibration of the shafting; a timing mark in fifths of a second; and the number of shaft revolutions.

Each instrument was provided with a light pulley, driven by a special fabric belt connected to the main shafting. In operation this pulley follows the torsional vibration of the rotating shaft. Loosely coupled to the light pulley and an integral part of the instrument is a heavy inertia mass of flywheel which rotates at a uniform speed. The relative motion between the light pulley and the inertia mass, through a linkage, actuates the pen of the instrument across the moving strip of paper. Through a medium of a pen, a clock run by battery records time on the tape in fifths of a second. The number of shaft revolutions are reported by electric impulses, relayed from a trip, set in coordination with the ship's revolution counter.

Sea trials were conducted on July 23 in the channel between San Pedro and Santa Catalina Island, California. Tape recordings were taken at various speeds in light and loaded ship conditions. At the same time, indicator cards were taken so as to establish the i.h.p. of the engine at various speeds and load conditions and at the same time provide a means for power balancing the engine. Ideal conditions as to wind and water prevailed throughout the sea trial and from preliminary calculation the test was successful. The engine was operated at 76 r.p.m. and there was no evidence of excessive torsional vibration. There was no indication of heating of No. 6 main bearing due to the increased bearing pressure from the added weight of the flywheel. Final information regarding these tests is being formulated by the American Bureau of Shipping and the U. S. Navy and will be available after all calculations are completed. Meanwhile it seems reasonable to assume from the trial observations mentioned that this interesting problem has been solved in a neat and not excessively costly manner.

# DISCUSSION.

By Mr. J. S. McGuire, *President, The Fairfax Engineering Company, Inc.*

ON

THE U. S. NAVY PLASTIC BOAT PROGRAM.

(page 16, Volume 61, Number 1, February 1949)

The instructive and informative article in the February issue of the JOURNAL on the "U. S. Navy Plastic Boat Program" by Commander A. C. Bushey, Jr., U.S.N.R. is in itself a refutation of claims that the Navy is reactionary in its outlook.

Commander Bushey's article presents an informed and critical evaluation of the progress made to date, and bears evidence to the farsightedness of the Navy Department toward the development, not only of an alternative material, but a superior material for Navy requirement.

The present development of high strength plastics reinforced by inorganic glass fibres may well be compared to the state of the radio industry prior to the development of the DeForest Tube. It seems that the initial development of any great new idea does fall into the hands of relatively inexperienced people. This is naturally to be expected, as those who are experienced and competent are too occupied with their own affairs to devote endlessly, of time and finance, toward the perfection of something which is visionary.

The counsel and coordination, as well as the financial support given by Research and Development Contracts by the Bureau of Ships, has had a great influence towards stabilization and development of the use of glass reinforced plastics for boat hulls.

Having been fortunate in having relatively long association with products where the "plus" factors of glass reinforced plastics made them mandatory rather than experimental, we have come to a number of specific conclusions which our experience has indicated is necessary to a uniform high quality production.

In the construction of hulls or super-structures for boats, and equally true of any other end product, we have two primary and vital considerations to contend with at one and the same time. Namely, we create the form and finish of the completed article, roughly paralleling any other type of manufacture. Second and most important, we create at the same time the basic material from which the end product is made.

It is therefore obvious that equal or more attention should be given to absolute correctness of procedure and a system of control that maintains absolute uniformity of that procedure.

The following comparisons are reasonably applicable: to produce the completed steel hull in the blast furnace where the ore is reduced or to complete a plywood boat at the saw where the logs are first slabbed.

From the time of application of the raw material, i.e. glass fibres and resins, to the mould nothing but process control can insure a uniform high quality.

The microscopic fine cylindrical fibres of glass represent an entirely different moulding technique than that required for any organic reinforcement. First, the smooth cylindrical fibres do not offer a mechanical bond as is found in reinforcing steel used with concrete. At the present time at least the material strength is largely determined by the density of the matrix or resins and the efficiency by which stress is passed through the matrix to the reinforcement itself.

The absorption of water or other fluids apparently has a lubricating action, but in some instances results show extreme loss of strength after submergence for a reasonable period. This is directly opposed to the use of organic fibres which by osmosis or capillarity tend to swell and increase the tightness within the matrices.

Our chemical companies are aware of this problem. Considerable research has been under way and is continuing toward a more perfect bond between the matrix and the fibres. The desirability of this is evidenced by the extreme strengths possible with properly oriented fibres. We have in many thousands of instances produced production mouldings of oriented fibres wherein the "tension in flexure" tests indicated the development of 80% of the theoretical strength of the fibre content.

In the use of random fibre, whether in the form of prepared "Mat" or suction induced preforms, reasonable goal should approximate the average of the sheer and tension developed by the oriented fibre specimens.

To revert to an earlier comparison, the development of resins and the art of moulding of glass reinforced plastics is only in the embryonic state. As security regulations do not permit an open, comprehensive evaluation of the entire problem, the responsible officers must confine themselves to end products. Due to the utter lack of experience in large mouldings by other than a very few companies, it has placed the Bureau in the apparent position of competitive bidding for "Know How," whereas the very existence of any "Know How" would preclude competitive bidding.

The necessity of encompassing the broadest possible experience, consistent with reasonable expectation of success, has been responsible for the spread of R. & D. Contracts to as many sources as possible. The net result has been that the tooling requirements must be amortized over relatively low production. To the writer's knowledge the Army's engineers have placed the largest quantity-bulk contract of any of the services, namely 60 units of approximately 200 cubic feet displacement each.

The development of glass and of resins is definitely moving in the research departments of the companies producing these materials. Commander Bushey touched upon, but did not elaborate on one other equally potent factor, namely equipment. It is generally recognized that maintaining a proper glass resin ratio for a given anticipated stress is essential. When large areas are involved and pressures upwards of 100 lbs. per square inch used in moulding, the clamping pressure used is tremendous. Hydraulic equipment on a scale never before visualized in American industry must necessarily be developed for positive control to produce the optimum results.

Intricate heat calculations coincident with the exothermic polymerization must be provided.

The development of composite designs where tensions are supported by oriented fibres and compressions by box, or compartment construction are all in the immediate future.

Although the activity of the Bureau of Ships in fostering research and development of plastics boats is confined to the end products necessary to the Navy, the information gained is a valuable contribution toward the production of other large area equipment. Railroad passenger cars might well be moulded. Truck bodies, aircraft wings, oil field tanks and piping, chemical process equipment and many other fields of activity are close to realization.

# BOOK REVIEW.

RESISTANCE, PROPULSION AND STEERING OF SHIPS, by Dr. Ir. W. P. A. Van Lammeren, Prof. Ir. L. Troost, and Ir. J. G. Koning. The Technical Publishing Company, H. Stam, Harlem, Holland, 1948, 366 pages, 240 figures.

Reviewed by: Captain Harold E. Saunders, U. S. Navy.

For a people who inhabit so small a country, the Dutch have large ideas. As one of the world's leading maritime nations they, like the Danes, have a small area from which to sail but a world-wide area in which to cruise. Their naval architects and marine engineers, undaunted by the vicissitudes of the recent war, have now embarked on a program of scientific and engineering publications which would do credit to any country, no matter how large.

The first book of this new series, which is to contain six volumes covering all phases of the design and building of ships and their machinery, has now appeared in an English version. It is well described by its title, and is the work of the Superintendent and the staff of the Netherlands Model Basin at Wageningen, principally Dr. Ir. W. P. A. Van Lammeren.

This book, in its original Dutch version, first appeared in Amsterdam in 1942. It was not made known to members of the profession at large until 1945, when a few copies were brought to this country. It was translated into English by the Dutch authors, and edited by Dr. F. H. Todd, then of the Ship Division of the National Physical Laboratory at Teddington, England. It has benefited by the inclusion of additional material since the publication of the first edition in Dutch seven years ago.

The book is divided into four main parts. The two principal parts cover the large subjects of Ship Resistance and Propulsion; the two minor ones deal with Steering and Ship Trials. Each of the first three parts begins with a rather extensive discussion of the theory involved, which is then followed by a description of tests and experiments with model and full-scale ships and propellers, which corroborate or modify the theories given. These parts then conclude with extensive sections devoted to the design of ship hulls, appendages, propellers, and rudders. The last and smallest part of the book, devoted to ship trials, describes a number of instruments for recording full-scale data which are not too well known in this country.

Rather more than half the pages of the volume are devoted to the treatment of a wide variety of aspects of the theory, performance, and design of the screw propeller. The inclusion of many design charts in the text acts to swell this part of the book. The space allotted to the treatment of propulsion problems has possibly limited the discussion of hull design to a few principal types, principally seagoing cargo and passenger ships, coasters, and river and canal vessels. These are the types, of course, of greatest interest to Dutch naval architects and marine engineers.

The book contains much material that is new in publications of this kind, relating especially to developments in the theory of water flow around ships and propellers during the past two decades, and the application of this new knowledge to technical and practical problems. It is not to be expected, however, that some of this knowledge will be come by easily in a reading of the book, as certain sections presuppose a degree of understanding that is none too commonly encountered, even among model basin personnel.

The book is generously illustrated with excellent drawings but one could wish for a few more of them, especially to explain the elementary nature of some of the hydrodynamic phenomena described. There are included line drawings of a large variety of ship forms and types, and the numerous types of modern ship rudders are rather extensively and elaborately depicted.

It is unfortunate that many of the symbols used are not well-known on this side of the Atlantic, but this is understandable. The format of the book is pleasing, and the type design employed for the numerals is superb. A somewhat larger and more legible size of lower-case subscript would have rounded off the format to perfection

This is a book deserving of intensive study on the part of every naval architect and propulsion engineer who wishes to familiarize himself with the latest and most useful developments in his profession.

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THE RADIO AMATEUR'S HANDBOOK BY THE HEADQUARTERS' STAFF OF THE AMERICAN RADIO RELAY LEAGUE. Published (1949) by the American Radio Relay League, Inc., West Hartford, Conn. The book is printed on 6½" x 9½" sheets and contains 736 pages. There are 1,651 illustrations and 118 charts and tables. The price is $2.00 in the United States and its possessions.

REVIEWED BY: LT. COM. C. W. HARRISON, JR., U.S.N.

Although the Radio Amateur's Handbook is written primarily for the person who wishes to get started and advance in the fascinating hobby of amateur radio, it has been favorably received by electronics engineers who desire a convenient and current reference for use in solving practical problems of electronics falling within the scope of the text. No one interested in the field of electronics should be without a copy of the Handbook.

There are 25 chapters devoted to a discussion of a variety of topics related to amateur communications, beginning with a discussion of the fundamentals of electrical circuits and vacuum tubes and ending with a most excellent compilation (in table form) of the characteristics of vacuum tubes, including magnetrons, klystrons, cathode-ray tubes and kinescopes. The information in the tube tables alone is worth the price of the book.

The 1949 edition of the Radio Amateur's Handbook is characterized by inclusion of the latest practical information on such subjects as single-sideband telephony, improved audio noise limiters, and band-switching preselectors. The wide fields of amateur v-h-f, u-h-f and microwave techniques are well covered, and much constructional data is included. Earlier editions of the Handbook contained information on the elimination of interference to broadcast receivers. The present edition contains information pertinent to reducing or eliminating amateur station interference with television, as well as broadcast reception.

In the antenna and transmission line section of the book, beginning on page 335, a number of minor errors of statement appear. For example, in the discussion of antenna impedance mention is made of the impedance of an antenna at any point along its length. If impedance is defined in the conventional way this is a meaningless concept, as anyone who has studied asymmetrical antennas will verify. The impedance of an antenna, on which standing waves exist, is generally not a pure resistance at current loops and nodes in contradiction to a statement appearing in the book.

To be technically precise one should speak of "charge distribution" instead of "voltage distribution" along an antenna comparable to the wavelength in overall length. The introduction of a per unit capacitance, and a per unit inductance for antennas is unfortunate. These concepts more properly belong to the theory of transmission lines. It is usually as easy to state a physical concept correctly as it is to state it in some arbitrary physically meaningless way and the chances of being understood, even by a beginner, are often improved.

One gains the impression from the book that the magnitude of a given impedance is all that counts in amateur radio. Actually it is just as important to know the phase angle as it is to know the magnitude of the impedance when it comes to impedance matching. Many amateurs as well as self-styled engineers are glibly unaware that such a thing as a complex impedance exists.

Basically the Radio Amateur's Handbook highlights the techniques employed to reduce to practice various phases of the communications art; that is, it translates complicated concepts and theories into practical circuitry. In spite of its minor faults the manual enables a specialized practicing engineer to keep a proper perspective in the ever-enlarging field of transmission and reception of radio signals. For persons who desire to acquaint themselves with the latest communication techniques, from the practical point of view, the book is indispensable.

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ATOMIC ENERGY DEVELOPMENT 1947-1948. Published by the U. S. Atomic Energy Commission and Printed by the U. S. Government Printing Office. Price 45 cents.

This fifth Semiannual Report of the Atomic Energy Commission to the Congress is interesting and informative reading. In its 213 pages it presents an excellent picture of the non-classified status of atomic energy development

The Chapter titles are:

I. Production.
II. Military Application.
III. Research in the Physical Sciences.
IV. Biology and Medicine.
V. Public and Technical Information Service.
VI. Security.
VII. Finance.
VIII. Organization and Personnel.
IX. Patents and Inventions.

The pamphlet is recommended for its easy readability. Its contents should be known as general information to every citizen.

## DEATH NOTICES.

The Society has learned with regret of the deaths of the following members. Time and facilities have prevented the preparation of individual obituary notices.

CIVIL MEMBERS.

A. F. E. Horn.

R. V. Terry.

## CHANGES IN MEMBERSHIP.

As of 1 February 1949, a complete list of members, of record on that date was published and distributed to all members. Since the list appeared, we have been notified of many modifications in titles, positions, and addresses. It is impracticable to publish all of these here. An early reissue is contemplated and each member will be given an opportunity to furnish the entry which he wishes to have printed in the list.

Since 1 February 1949, the following new members have been accepted into the Society:

### NEW MEMBERS.

#### NAVAL.

Anderson, Alonzo G., Ch. Mach., U.S.N.,
U.S.S. *Metaco* (ATF 86) c/o FPO, San Francisco, Calif.
Borop, James D. W., Lieut. Commander, U.S.N.,
Staff, Com. Crulant, U. S. Naval Base, Norfolk 11, Va.
Box, Claude D., Ensign U.S.N.R.,
Room 3318 Cleveland Hall, U. S. M.M. Academy, Kings Point, N. Y.
Bradford, Guy William, Jr., Ensign, U.S.N.R.,
Room 3417 Cleveland Hall, U.S.M.M. Academy, Kings Point, N. Y.
Briggs, Foster LeRoy, Lieutenant, U.S.N.,
2951 N. Front St., Harrisburg, Pa.
Britt, James Jr., Lieutenant U.S.N.R.,
U.S.S. LST 519, c/o FPO. New York, N. Y.
Burke, Harold J., Captain. U.S.N.R., Manager Systems Div.,
CO Two Fire Equipment Co., U. S. Highway No. 1, Newark, N. J.
Caldwell, Clarence Mills, Commander, U.S.N.,
Deputy, Contract Div. Bureau of Ships, Navy Dept.
Mail: 5511 East Glen Lane, Chevy Chase 15, Md.
Cestone, Joseph Anthony, Lieutenant, U.S.N.R.,
In charge Gyro Compass Section, I. C. & F. Branch, Bureau of Ships, Navy Dept.,
Mail: 3420 39th St., N.W., Washington 16, D. C.
Cray, Edward P., Lieutenant, U.S.N.R., Sales Engineer, Gulf Oil Corp.
Mail: 8 Gertrude Road, West Roxbury, Mass.
Culver, John Arthur, Ensign, U.S.N.R.,
U.S.S. *Weiss* (APD 135) c/o FPO, Norfolk, Va.
Danby, Leo Chandler, Chf. Radio Elect., U.S.N., Bureau of Ships, Navy Dept.
Mail: 1409 48th Ave., Washington 19, D. C.
Darden, Arthur Dandridge, Lieutenant j.g., U.S.N.R.,
Asst. Engr. Welding Shipyards, Inc.
Mail: 4100 Warren St., Apt. 2, Norfolk 5, Va.
Davidson, John D., Ensign, U.S.N.,
U.S.S. PCE 904, c/o FPO, San Francisco, Calif.
Davis, Wallace S., Lieutenant, U.S.N.R.,
Rheem Mfg. Co.,
Mail: 2403 Lee Boulevard, Arlington, Va.

Desel, Robert Frederick Paul, Lieutenant, U.S.N.,
David Taylor Model Basin, Washington 7, D. C.

Eaton, William Arthur, Commander, U.S.N.R., Ret.,
Resident Manager, Raytheon Mfg. Co., Waltham, Mass.

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U. S. Maritime Service
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Tullier, P. M., Jr., Lieutenant, U.S.N.R.

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Harmon, R. N.
Jackson, J. B.
Lorraine, R. G.

ASSOCIATE.

Chamberlin, H. B.
Lubbers, A. W.
Miller, Carlyle W.

# ASSOCIATION NOTES.

## Original Articles.

The editor of the Journal is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $50.00 to $150.00 depending upon length, interest and professional value.

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. Include on a separate page a short biographical sketch(es) of the author(s). 50 to 100 words for each author is desired.

Manuscripts should be mailed addressed to:

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Bureau of Ships, Navy Department
Washington 25, D. C.

Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 10 reprint copies of his article free of charge. Additional copies may be purchased from the Society if the request is received 30 days prior to the publication date which is the 25th of the issue month. Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Manuscripts which are accepted will be published in the earliest Journal which has not yet been closed (60 days before publication) and for which insufficient material is already on hand.

## Subject Matter and Authors.

To assist the editor in programming future issues, the help of all members is solicited. To make it easy, the last page in this issue is readily detachable as you will find. This page has blanks for two entries.

(*a*) What subject would you, as a member, like to see covered by an article in the Journal.

and

(*b*) Who do you consider best qualified to prepare such an article authoritatively, and what is his address. (if you know).

Any of these forms which are filled in and mailed to the Society will be received with thanks and the editor will follow up to get the suggested articles prepared.

## ASSOCIATION NOTES.

### PHOTOGRAPHS.

Photographs of current or historical interest to readers of the Journal are desired. Any which are accepted by the Society will be purchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

### PRIZE ESSAY CONTEST.

The 1949 Prize Essay Contest was very disappointing. Very little interest evidenced if the number of entries submitted may be taken as a criterion. No award was made.

### ANNUAL BANQUET.

The Society's Annual Banquet was held at the Statler Hotel in Washington, D. C. on April 29th. In many respects it was the most successful of a long series of these delightful affairs. The attendance badly strained the capacity of the facilities. Over a hundred members and guests more than attended the 1948 banquet were served.

Rear Admiral Thorwald A. Solberg, U. S. Navy, President of the Society presided. Vice Admiral Earle W. Mills, U. S. Navy, Retired, a past-President, officiated as toastmaster.

The assembly was treated to two outstanding addresses. Mr. Charles E. Wilson, President of the General Electric Company discussed "D-Day on Market Street." Vice Admiral Robert A. Carney, U. S. Navy, Deputy Chief of Naval Operations for Logistics chose as a subject: "The impact of Unification on Naval Engineering."

Those attending were favored throughout the banquet with musical selections rendered by the United States Marine Band under the direction of Major William F. Santelmann, U.S.M.C.

### NOTICE OF DEATHS.

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be appreciated greatly if any member who learns of the death of a member will advise the Secretary-Treasurer.

### ADDRESSES.

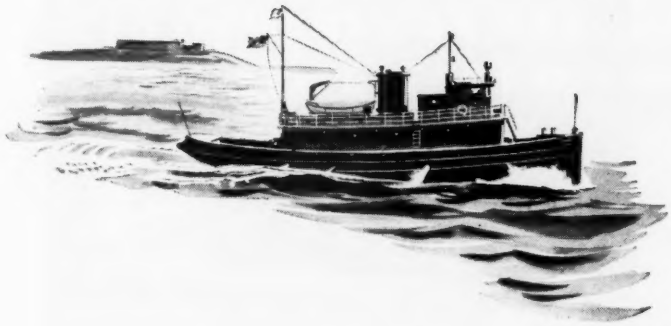
Once more attention is invited to the necessity of keeping the Society advised as to changes in addresses.

### SOCIETY LAPEL BUTTON.

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold back-ground.

It is available to all members at fifty cents (50c) each.



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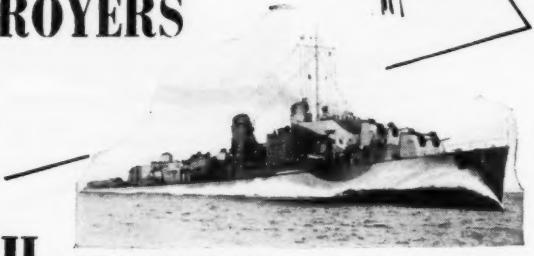

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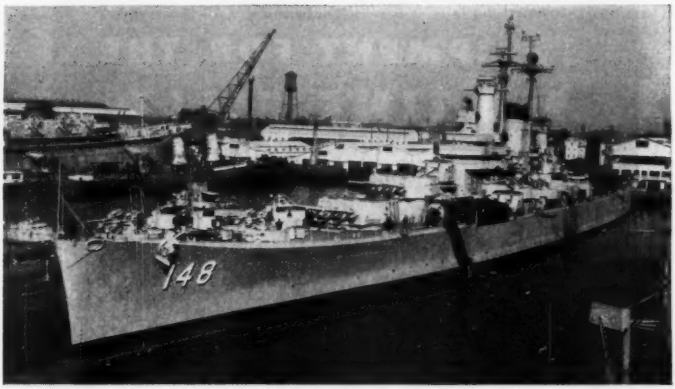


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The U.S.S. "NEWPORT NEWS," a heavy cruiser of the Salem Class is shown at Newport News on January 29, 1949, during her commissioning ceremony. Of the many Navy vessels constructed at Newport News, she is the first to bear the name of that shipbuilding city.

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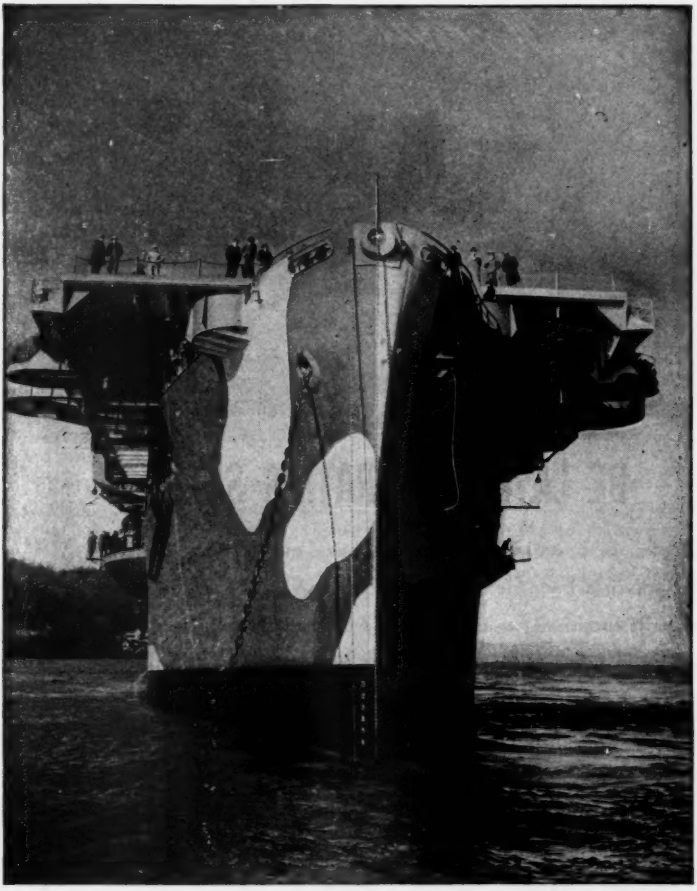

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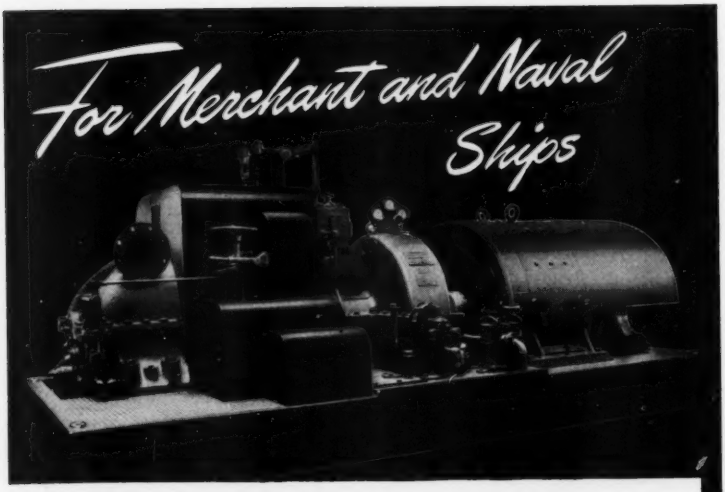


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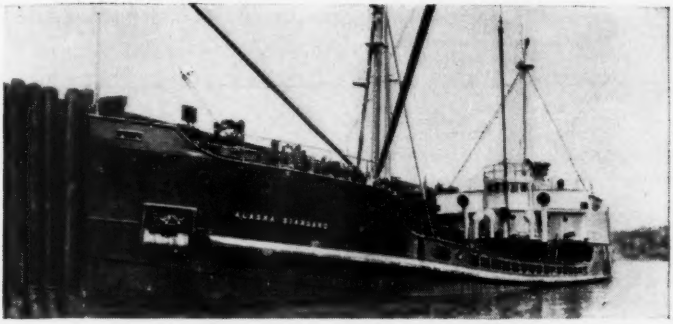
# G-E propulsion drive in service 24 years without repairs



A fine example of the trouble-free durability of G-E marine propulsion drives is the case of the 210-foot tanker MS *Alaska Standard*. For 24 years this "floating bulk plant" has supplied petroleum products to fishermen, fur trappers, and storekeepers along the southeastern Alaska coast. Although it has logged close to a million miles, not once have major repairs been needed on her G-E diesel-electric drive!

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Model 37: 500 to 2125 horsepower for direct drive marine service.
Model 42: 60 and 90 horsepower at 450 r.p.m.
Model 31: Horsepower from 175 to 500; with 2 to 1 reduction gears optional for marine service.
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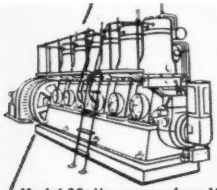

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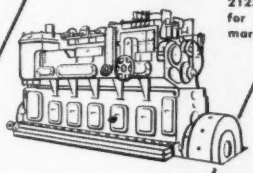

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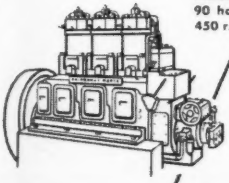
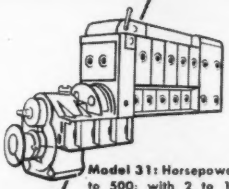
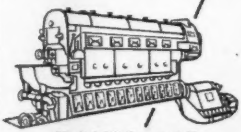


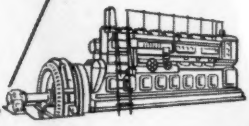


Model 31: Horsepower from 175 to 500; with 2 to 1 reduction gears optional for marine service.



Model 38 Opposed-Piston Engine: Horsepower from 960 to 1920 with 2 to 1 reduction gear for marine service.

Model 33 Stationary Engine: 500 to 2000 hp. Diesel, 1000 to 2000 hp. Dual Fuel.



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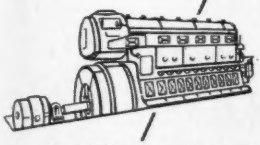
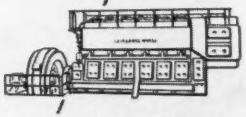


Model 38 Opposed-Piston Engine: 960 to 1920 horsepower at 720 r.p.m.



Model 31: 2100 to 3500 horsepower diesel or dual fuel engine for heavy-duty stationary service.

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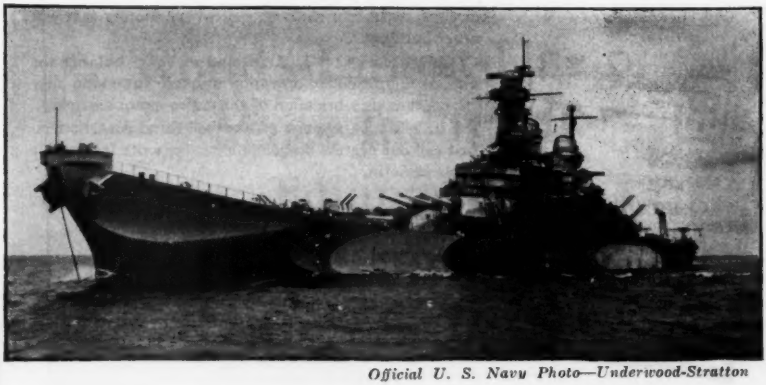

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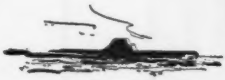

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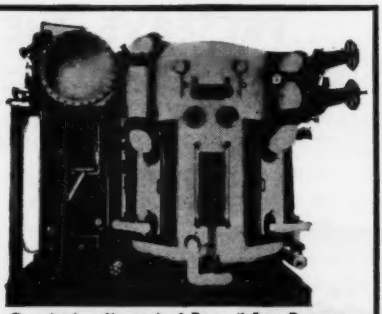

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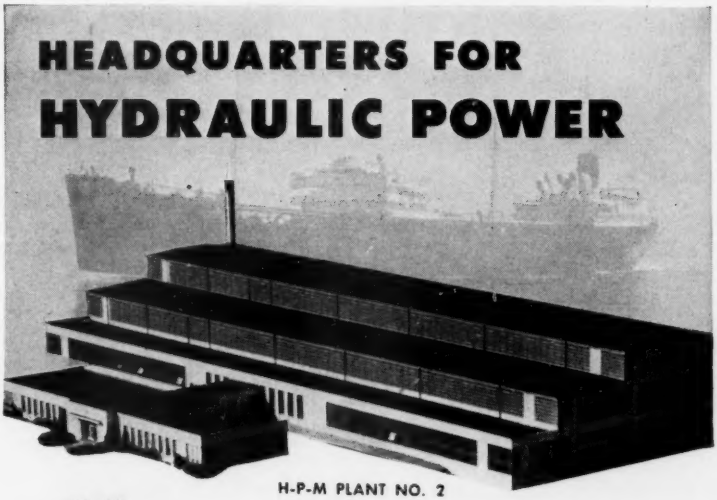

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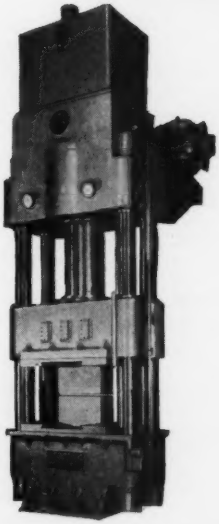

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.....

I suggest that.....

.....could prepare an authoritative article on the above subject.

.....
Member

## QUALIFICATION FOR MEMBERSHIP

Commissioned and ex-commissioned officers of the regular Navy, Marine Corps and Coast Guard of the United States; warrant and ex-warrant officers of the regular Navy, Coast Guard and Marine Corps of the United States; reserve commissioned and warrant officers of the Navy, Coast Guard and Marine Corps of the United States shall be eligible as Naval Members. Persons eligible as naval members shall be admitted upon application and payment of annual dues.

Persons in civil life whose knowledge of engineering is such that they can cooperate with Naval engineers in the promotion of professional knowledge may be eligible as civil members. They shall have been in the active practice of an engineering profession for at least eight years and in responsible charge of important work for five years, and shall be qualified to design as well as to direct engineering work. Fulfilling the duties of a professor of engineering who is in charge of a department in a college or school of accepted standing shall be taken as an equivalent to an equal number of years of active practice. Graduation from a school of engineering of recognized standing shall be considered as equivalent to two years of active practice. Persons eligible as civil members may be admitted upon application and payment of annual dues, provided that the application is accompanied by the recommendation of two members and provided that the application shall receive the approval of a majority of the Council.

Persons in civil life who are not eligible for civil membership, but who are especially interested in naval matters or the merchant marine may be eligible as associate members. Commissioned officers of the United States Army and of foreign military and naval services may be eligible as associate members. Persons eligible to associate membership may be admitted upon application and payment of annual dues, provided the application have the recommendation of a member and provided the application shall receive the approval of a majority of the Council, except that in the case of commissioned officers of the United States Army and of foreign naval and military services, the recommendation of a member will not be required.

Associate members shall be entitled to all the privileges of other members except voting and holding office.

The annual dues shall be $5.00, payable on 1 January in advance.

## APPLICATION FOR MEMBERSHIP IN THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

Date .....

I hereby make application for membership* in the American Society of Naval Engineers and submit the following information:

### For Naval Membership

(First) (Middle) (Last)

Name .....

Rank ..... File No. .....

Business connection and position, if any .....

### For Civil Membership

(First) (Middle) (Last)

Name .....

Years in engineering work .....

Years in responsible charge of important work .....

Present business connection and position .....

Recommended by (two members) .....

### For Associate Membership

(First) (Middle) (Last)

Name .....

Rank, if Commissioned Officer of U. S. Army or of foreign military or Naval service .....

Business connection and position .....

Recommended by (one member) .....

.....

*Signature of Applicant*

Address for Journal and Mail

.....

.....

MAIL TO SECRETARY-TREASURER
THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.
BUREAU OF SHIPS, NAVY DEPT., WASHINGTON, D. C.

* See reverse side for required qualifications for various classes of membership.

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Captain J. E. HAMILTON, U. S. Navy

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